THE FEASIBILITY OF HEAT PUMPS FOR DOMESTIC HEATING IN TASMANIA

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Cover: 2 kW heat pump developed by the Building Research Establishment in Britain.

CONTENTS

GLOSSARY

Auxiliary heating: resistance heating used in conjunction with a heat pump, to boost heat output *above the rated heat pump output* under high load conditions.

η mechanical efficiency.

A_{Du}: Du Bois Area: effective area for heat loss from the human body (section 3.2).

A.B.S.: Australian Bureau of Statistics.

A.I.R.A.H.: Australian Institute of Refrigeration and Heating.

A.S.H.R.A.E.: American Society of Heating, Refrigeration and Air-conditioning Engineers.

Absorption heat pump: a heat pump that operates from a heat source, rather than a mechanical compressor (Figure 2.3).

Activity level: rate of heat production in the human body (section 3.2).

Ambient temperature: effective temperature for human thermal comfort (section 3.2).

Building envelope: walls, windows, floor and roof.

COP: Coefficient of Performance; measure of the efficiency of a heat pump (section 1.2).

C.P.I.: Consumer Price Index.

Changeover system: zoned, ducted heating system which heats the living area of a house by day and the sleeping area by night.

Climatic adjustment factor: parameter derived to compare the economics of heat pump installations at different locations (section 3.4).

clo-value (clo): dimensionless term measuring thermal resistance of clothing (section 3.2).

Condenser: section of a heat pump which gives out heat (section 2.2).

d.d. (Degree day): measure of the severity of climate with respect to annual heating requirements (section 3.4).

Design temperature: outdoor temperature used for selection of required heating capacity (section 3.4).

Direct electric heating: electrical resistance heating operating on the household tariff.

E.E.R.: Energy Effectiveness Ratio (section 1.2).

Evaporator: section of a heat pump which absorbs heat (section 2.2).

GJ (Gigajoule): 10^9 Joules, or 1000 megajoules (MJ).

Heat bank: insulated electric storage heater using a fan to bring out the stored heat as required.

Heat storage: storage of heat (as raised temperature) in the thermal mass of a heat store, or in the structure of a building.

High grade energy: energy which can be efficiently converted to mechanical work.

High grade heat: heat ($\gtrsim 100^{\circ}$C) which can be efficiently converted to mechanical work.

I_{cl}: clo-value.

Initial cost: combined costs of purchase, installation and accessories.

Internal admittance: thermal storage capacity of a building element (section 4.2).

Internal heat gains: heat from occupancy, lights and appliances, which supplies part of the total heating load.

J: joule.

kcal: kilocalorie (= 4,186.8 J).

kW: kilowatt (= 10^3 watts).

kWh: kilowatt hour (= 3.6 MJ).

Lag: thermal response time of a building element (section 4.2).

Lead: thermal response time of a building element (section 4.2).

Low grade energy: energy which can be converted to mechanical work only at low ($\lesssim$ 70%) efficiency.

Low grade heat: heat at low temperatures ($\lesssim 100^{\circ}$C) which cannot be efficiently converted into mechanical work.

M: metabolic rate.

m: metre.

MJ: Megajoule (= 10^6 joules).

Mean radiant temperature: defined as "that uniform temperature of black surroundings producing the same net radiant heat exchange as the surroundings being considered".

Mechanical heat pump: heat pump operating on the vapour compression cycle (section 2.2).

Midi-bank: passive electric storage heater which emits heat continuously.

NPC: Net Present Cost (section 6.2).

PMV: Predicted Mean Vote (for thermal comfort; section 3.2).

PPD: Predicted Proportion Dissatisfied (section 3.2).

R: thermal resistance of a building element.

RFL: Reflective Foil Laminate ("Sisalation").

Refrigerating COP: measure of the efficiency of a refrigerator or air-conditioner (section 1.2).

Seasonal COP: ratio of heat produced to energy consumed, averaged over a heating season (including supplementary resistance heating).

Single package heat pump: heat pump incorporating compressor, evaporator, condenser and fans in a single unit.

Skirting heater: passive electrical convection heater mounted as a skirting board.

Split system heat pump: heat pump with the compressor and evaporator separated from the other components.

Split system console heat pump: split system heat pump distributing its heat from an indoor heating console, rather than through ducting.

Supplementary heating: direct electric heating used to supplement the heat output of a heat pump or electric storage heater under conditions of low temperature and/or high load.

Sol-air temperature: the (fictitious) air temperature at which the same rate of heat loss would occur through a building element in the absence of all radiation effects.

t_a: air temperature.

t_{mrt}: mean radiant temperature.

Thermal comfort: "that condition of mind which expresses satisfaction with the thermal environment".

Thermal storage: see "heat storage".

Through-the-wall heat pump: single package heat pump (or air-conditioner) mounted through an external wall, without air ducting.

Transfer modulus: thermal storage capacity of a building element (section 4.2).

U-value: measure of the rate of heat loss through a building element ($U = {}^1/R$).

Vapour compression cycle heat pump: see "Mechanical heat pump".

Watt: one joule per second.

CHAPTER ONE: INTRODUCTION; ENERGY CONCEPTS

1.1 INTRODUCTION

From 1950 to 1975, Tasmania's rate of energy consumption - particularly consumption of petroleum products - increased considerably. This increase was due to a seemingly unquestioning acceptance of relatively cheap energy, in forms which could supply the real and apparent needs of people and industries.

Since the oil crises of the seventies, cheap and abundant energy supplies are no longer taken for granted. On present trends, new electricity sources will be needed, by the end of the century, to supplement Tasmania's hydro-electricity. Some means of replacing petroleum as a fuel for transport will be needed as oil reserves are used up.

In the medium term, the increasing cost of oil is expected to be the biggest problem. Many current uses of oil will become uneconomic. Without adequate planning, oil-using equipment may have to be scrapped - at considerable expense - as it becomes simply too expensive to operate. Almost all aspects of modern life, including many now taken for granted, will be affected.

The impact of the expected oil shortage can be reduced by conserving energy in general, and - wherever and whenever possible - substituting alternative forms of energy. In Tasmania, that means hydro-electric power, coal, wood, natural gas and solar-based energy sources. It should be borne in mind that this will place greater demands on *all* energy sources. Only the most efficient processes, with the least wastage, should be encouraged.

This study arises out of a concern for Tasmania's ability to weather the storms of the coming oil shortage, and considers the prospects for energy conservation in domestic heating. Domestic heating currently accounts for one-eighth of Tasmania's total energy use, including 8 per cent of the consumption of petroleum products (Hartley, Jones and Badcock, 1978). In particular, the study considers the potential of *heat pumps* as an energy-saving alternative to conventional forms of heating.

What is a Heat Pump?

Broadly speaking, any device which moves heat energy from one place to another is a heat pump.

[Water, which naturally flows downhill - i.e. from "up" to "down" - can be pumped uphill - from "down" to "up". So heat, which naturally flows from *hot* to *cold* can be "pumped" from cold bodies to hotter ones.]

Australians are already familiar with heat pumps, though they know them by the names of *refrigerator* and *reverse-cycle air conditioner*.

The refrigerator pumps heat energy, from the food stored inside it, to the heat exchanging coil located at the rear. Only a very small amount of heat needs to be removed to keep the refrigerator cold. This heat can be felt as a gentle warmth in the coil when the motor is running.

The reverse-cycle air conditioner is normally used to pump heat from a room to the air outside. In its reverse (or heating) cycle, it pumps heat from the outside air into the room. It could just as well be called a reverse-cycle heat pump!

A heat pump for heating a home has a much larger capacity than a refrigerator, and usually provides more heat than a reverse-cycle air conditioner. It takes its heat from a source located outside the house. Air, water and earth are commonly used low-temperature sources of heat.

The advantage of a heat pump is its ability to pump *more* energy (as heat) into a house than it consumes in the act of pumping the heat. To put it another way, it takes less energy to pump a given amount of heat into a house (using a heat pump) than to use energy directly to produce heat.

In practice, the efficiency and heat output of a heat pump depend on a number of factors, including the temperature, heat source, and the size and type of heat pump used. Typically, a heat pump can provide two and a half units of heat for each unit of energy it consumes.

Since it can provide either heating or cooling - or even both, simultaneously, to different parts of a building - the heat pump is an ideal means of air-conditioning large office blocks. A number of office blocks in Tasmania use heat pumps. Included among them are the State Offices block, which uses an air-source heat pump, and the Hydro-Electric Commission building, which uses the water in the Derwent River as a source of heat for its heat pump.

A scheme, presently underway, will reticulate hot and cold water from a single large heat pump, to provide air-conditioning for several office blocks in Hobart. This is believed to be the first reticulated heat pump system in Australia.

Smaller heat pumps, designed for heating in single-family houses, have recently become popular in the United States, Europe and New Zealand. In Tasmania, they are virtually unknown, for two reasons: they have a high initial cost; and manufacturers have been reluctant to promote them in an uncertain market.

Heat pumps can, in theory, contribute to energy conservation. In practice, this can only happen if they gain market acceptance. For this to happen, it must first be shown that the high initial cost is more than compensated for by improved heating and reductions in running costs.

One of the primary aims of this study is to examine the balance between initial costs and running costs in order to show the conditions under which heat pumps are economically competitive with conventional heaters.

Thus it will attempt to *define* the potential market for heat pumps in Tasmania. In doing so, it will also serve as a tool, helping to *create* that market.

In the remainder of this first chapter, a number of terms relating to energy will be defined and explained, and the outline of an energy conservation programme, specifically for Tasmanian conditions, will be developed.

1.2 BASIC ENERGY CONCEPTS

In order to discuss the efficiency of heating devices, it is necessary first to recall some fundamental principles which relate to energy:

(i) Heat is a form of energy:
- Heat may, in principle, be converted to any other form of energy (e.g. sound, light, mechanical energy, electrical energy); conversely, other forms of energy may be converted to heat.

(ii) Heat energy is related to temperature:
- Temperature is a measure of the mean motion of the molecules of a substance. The motion of the molecules comes about as a result of the heat energy they possess. Usually, heat energy and temperature change in proportion to each other. However, when a change of state occurs, the amount of heat energy changes a certain amount without a corresponding change in temperature.
The relationship between heat energy and temperature for a typical substance (water) is shown in Figure 1.1. Of course, two units of a substance, at a given temperature, contain twice as much heat energy as one.

(iii) Heat naturally flows from a warmer body to a colder one:
- "You can't pass heat from a colder to a hotter
You can try it if you like, but you're far better not'ter"
(Flanders and Swann, 1957)

To obtain heating energy from a heat source at ambient temperature, it is necessary to use a heat pump to raise the temperature of the heat. The methods by which this can be done are discussed in Chapter Two.

(iv) Energy is never "used up", but merely changes its form:
- For example (see Figure 1.2), the electrical energy supplied to power an electric motor is mostly converted to mechanical energy. The remainder is converted to heat and sound. Eventually, the mechanical energy - through friction processes - is converted to heat energy. The end result is that the electrical energy originally supplied is almost completely converted to heat energy.

It will also be useful to distinguish between *high-grade* and *low-grade* energy sources:

High-grade energy sources are those which can be almost completely converted to useful *work*. They include electrical energy, mechanical energy and the gravitational energy of water (which can be efficiently converted to electricity, or directly into useful work).

Low-grade energy sources can be converted to useful work at efficiencies below about 70 per cent. Most low-grade energy sources are first converted to heat energy, which is then used to drive an engine. It is in the conversion of heat to mechanical energy that inevitable losses (in the form of heat) occur. Low-grade energy sources include wood, coal, oil, gas, nuclear and solar.

Heat sources may be further divided into *high-grade* and *low-grade*, according to the temperatures they can provide. Temperatures above $100^{\circ}C$ can provide steam for powering steam engines or turbines, and are obtained from *high-grade heat sources*. *Low-grade heat*, at temperatures below $100^{\circ}C$, is still potentially useful for space heating and hot water. Often, however, low-grade heat is treated as waste by industry. However, industries in Britain (Department of Energy, 1977) have demonstrated that investment to make use of low-grade heat can repay itself in as little as ten months. When there is a surplus of low-grade heat (e.g. in a thermal power station) energy can be conserved by reticulating the heat to nearby housing, in a *district heating* scheme.

Conventional heaters use high-grade heat (wood, oil and gas) or even high-grade energy (electric heaters) to produce low-grade heating. Solar heating and district heating use low-grade heat for heating. Heat pumps, which use high-grade heat or energy to pump low-grade heat, are intermediate between the two.

Efficiency

There are a number of ways of defining efficiency. The most widely accepted definition (at least in scientific circles) is *thermodynamic* (or *energy*) efficiency. Other measures which will be used in this

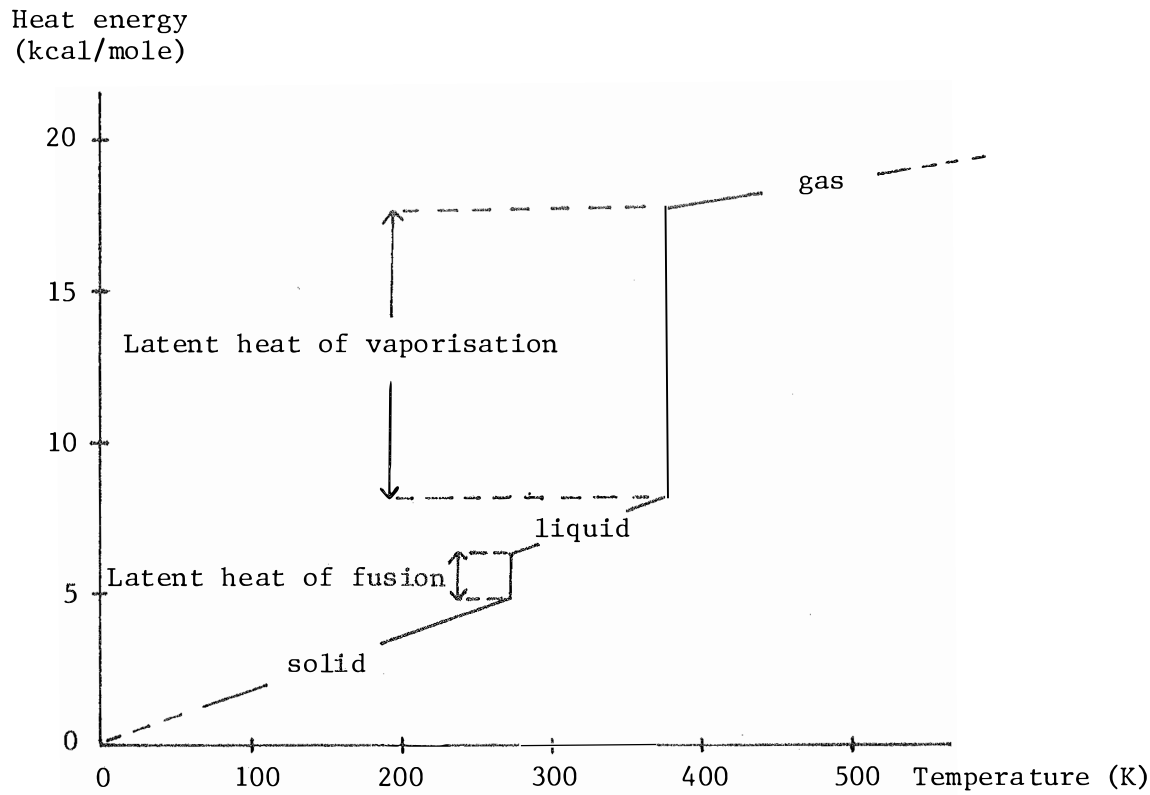


Figure 1.1: Relationship between heat and temperature (H_2O at 1 atm.).

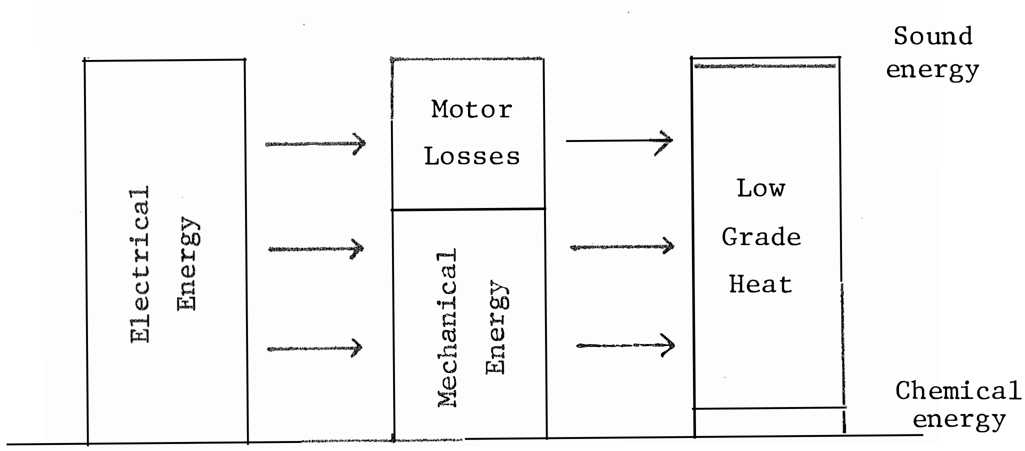


Figure 1.2: Typical flows of energy, using a kitchen food processor.

paper are *Energy Effectiveness Ratio* (EER), *Coefficient of Performance* (COP) and *Heating Effectiveness*. These terms will now be defined.

Thermodynamic Efficiency (η):

Thermodynamic efficiency is the proportion of the total energy consumed (in whatever form) which is converted to energy in the form desired. η may range from 0 to 1.0 (0–100 per cent).

Thus, when a 100-watt electrical appliance (e.g. a food processor) is said to be 60 per cent efficient, this means that it produces 60 per cent of 100 watts (i.e. 60 watts) of mechanical power in its blades. Of the remaining 40 watts, around one watt escapes as sound. The rest is converted to low-grade heat through processes such as friction in the motor. This low-grade heat is sufficient to make the motor housing warm to the touch after a few minutes of operation.

The high-grade mechanical energy of the blades becomes energy of motion of the food being processed. This energy of bulk motion is soon dissipated, either being converted to energy of motion of the individual molecules of the food (i.e. heat), or contributing to a change of state in the food (e.g. melting of ice or whipping of egg-whites).

This process is illustrated in Figure 1.2. The food processor, which has only a 60 per cent conversion efficiency of electrical to mechanical energy, converts electrical energy to heat with almost 100 per cent efficiency.

Coefficient of Performance (COP):

If an ordinary electrical appliance can heat with almost 100 per cent thermodynamic efficiency, how can a heat pump improve on that? The answer lies, not in its thermodynamic efficiency, but in its ability to use low-grade heat. Since the low-grade heat comes at no cost to the user, it is reasonable to omit the energy input of the low-grade heat from the calculation.

In such a case, the appropriate measure of performance is the coefficient of performance:

$$\text{COP} = \frac{\text{heating energy provided}}{\text{energy or high-grade heat input}}$$

(NOTE that this is not the same as the COP used to evaluate refrigeration.

In cooling, the high-grade energy input is rejected, but a well-designed heat pump can utilise this energy for heating, to increase its COP.)

The energy flows involved in electrical resistance and heat pump heating are compared in Figure 1.3, which illustrates the difference between thermodynamic efficiency and COP.

In conventional heaters, which use no low-grade heat, η = COP. When a heat pump is used to obtain low-grade heat, the COP rises according to the proportion of low-grade heat obtained for heating.

Energy Effectiveness Ratio (EER):

The two definitions above are measures of the performance of the *appliance* at the end of the energy chain. They take no account of the efficiency with which the electricity used is produced.

In most of the world, the choice between electricity and (for example) oil is a choice between two means of utilising the same fuel (oil). If the electricity is generated at 40 per cent efficiency from oil, then it takes less oil to fuel a 70 per cent efficient oil heater than to provide electricity for a 100 per cent efficient electric heater. In such a case, it is the efficiency of use of the *primary* energy resource that is important. This is measured by the Energy Effectiveness Ratio, defined as:

$$\text{EER} = \frac{\text{amount of energy produced in the desired form}}{\text{amount of primary energy consumed}}$$

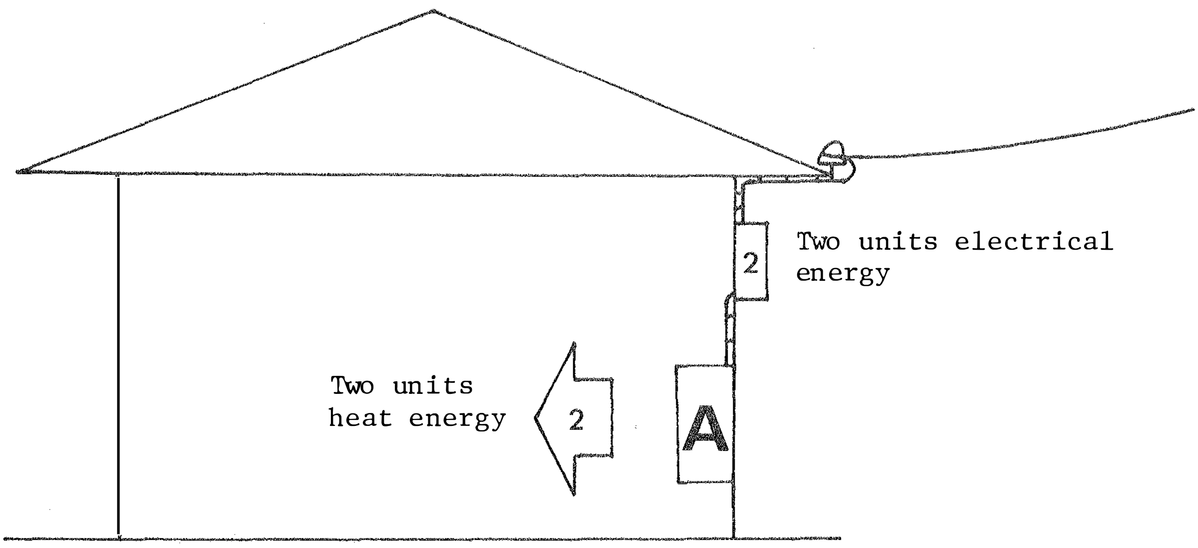
The EER takes account of the whole system involved in transforming the primary energy to its useful end.

Figure 1.4 depicts a number of means of obtaining heating from coal, with EER's varying from 0.2 to 1.25.

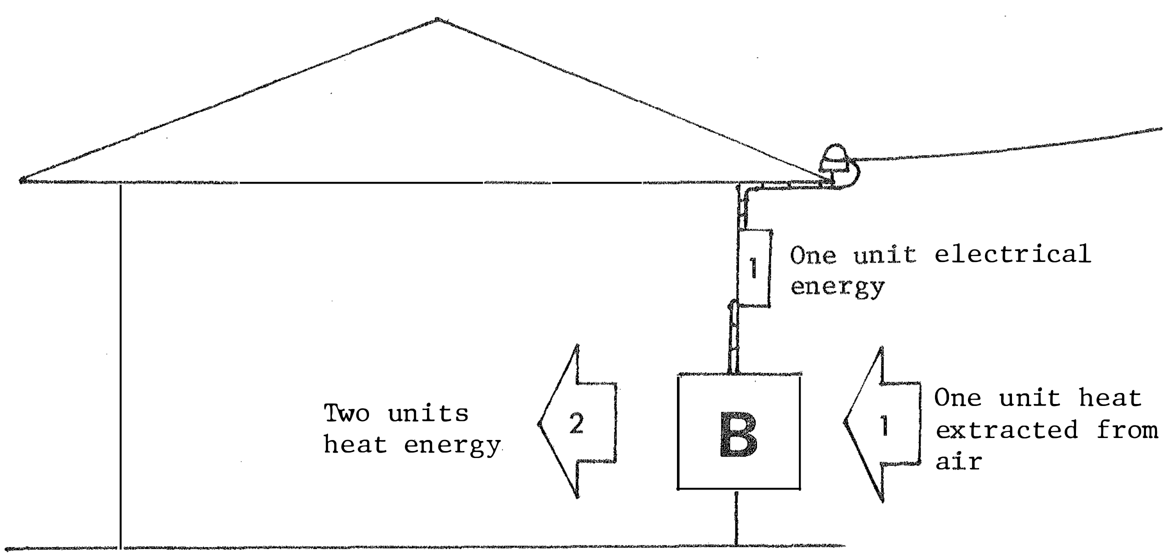
EER is among the best single measures for evaluating different means of using a limited energy resource. When comparing *different* primary energy sources and/or *different* end uses, the relative abundances of the energy sources and the relative demands on the end-uses are more important than the EER, which can then be used only as a rough guide.

Heating Effectiveness:

The ultimate aim of a heating system is not to provide heat energy in



$$\eta = \frac{2}{2} = 1.0$$



$$\eta = \frac{2}{1+1} = 1.0$$

$$COP = \frac{2}{1} = 2.0$$

Figure 1.3: Comparison of basic energy flows:
A: conventional electric heater
B: heat pump

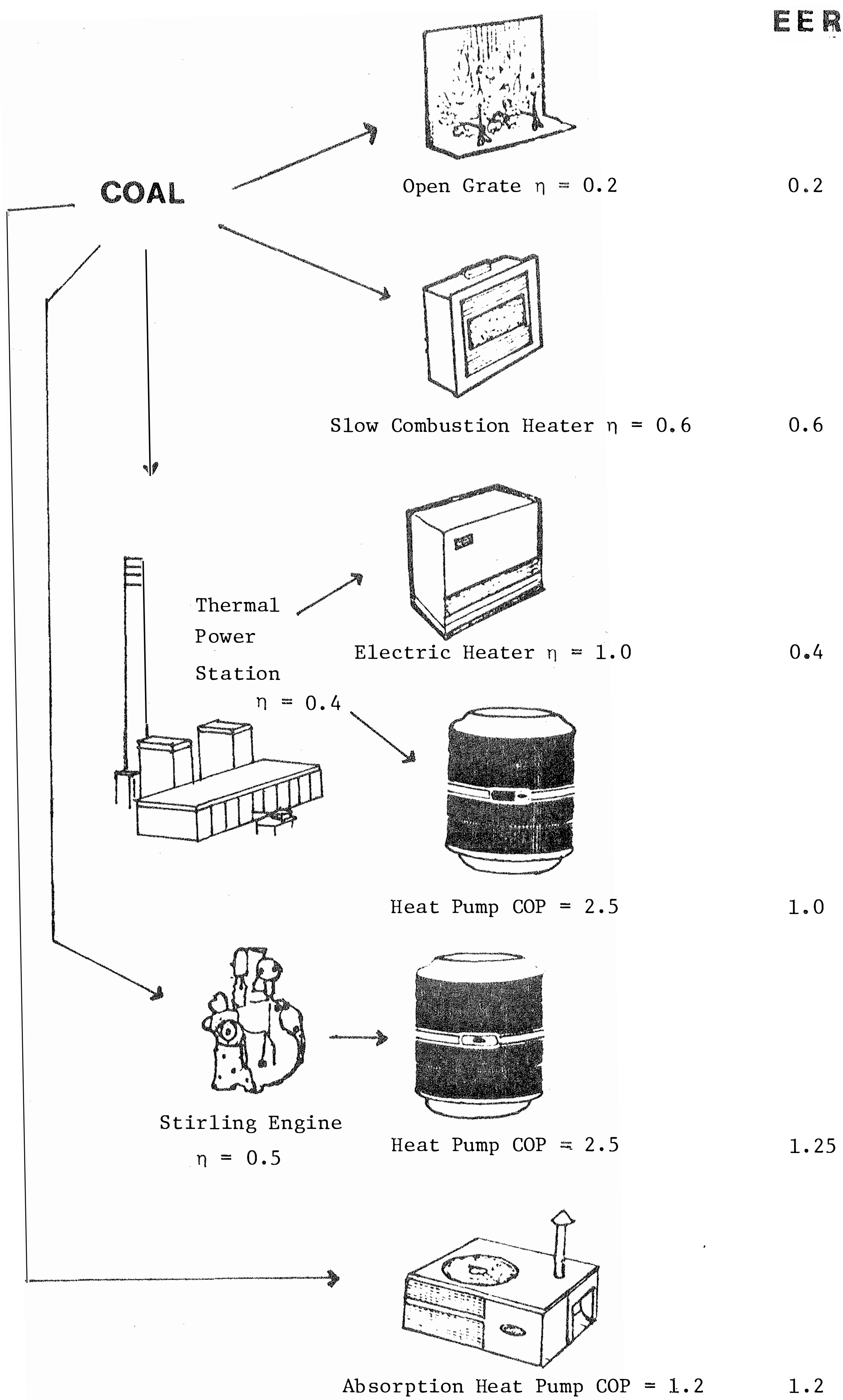


Figure 1.4: Energy efficiency ratios of different ways of obtaining heating from coal

the whole or part of a building, but to produce a feeling of *thermal comfort* in the occupants. In a given situation, different types of heater will produce more or less heat energy achieving thermal comfort, depending on their heating effectiveness.

Heating effectiveness is at present largely an intuitive concept. It will be further discussed in Chapter Two.

1.3 ENERGY CONSERVATION IN THE TASMANIAN CONTEXT

The most notable trends within our increasing consumption of energy are the decline of the use of firewood and associated greater use of petroleum products.

Tasmania's main indigenous energy resources are hydro-electric power, coal and wood. There is a small, untapped reserve of natural gas in the Pelican Field in Bass Strait. The potential contributions of solar, wind, tidal and wave power are as yet largely unknown, and are not expected to be large within the century.

Tasmania is unusual in having base-load electricity provided by hydro-electric power. Present capacity, plus the capacity of schemes currently under construction, will be adequate to fill the projected demand until 1986. With these schemes, around two-thirds of the hydro-electric potential will be exploited. Further development of hydro schemes is uncertain, due to environmental and economic factors. Hydro-electric power can be relied upon until the dams either silt up or become unsafe, in 100-300 years' time.

Coal reserves are estimated to last for at least a century at present (low) rates of consumption.

Wood, which was once a major energy source in Tasmania, has been largely replaced by oil in many applications. In recent years, wood has been viewed as an export commodity (as woodchips and paper pulp) rather than an energy source. In energy terms, the 1.7 million tonnes (*Tasmanian Year Book*, 1978) of exported woodchips would represent about 40 per cent of present energy consumption.

Surprisingly, Tasmania only produces 40 per cent of the energy it consumes. The rest is imported, mainly as petroleum products. As oil becomes more expensive and difficult to obtain, the major energy problem will be that of replacing oil - or learning to live without it!

The most difficult area will be that of transport, which has no real alternative to petroleum. With enormous capital investment, the State's rail system could be upgraded to supplement the existing road network. Trains could be run on electricity, wood or coal. To completely replace road transport with rail would place enormous strains on our energy resources. It is more likely that, as transport costs increase, the use of transport will diminish. Even if we can solve the transport problem, there will be difficulties in non-transport areas which currently obtain much of their energy needs by oil products.

As oil becomes uneconomic, people will switch to other energy sources to fill their needs. This in turn will strain supply. Without sufficient forward planning, demand for local energy sources - electricity, coal, wood - may unexpectedly exceed supply.

What is needed is a complete reappraisal of energy use in Tasmania, using the following guidelines:

(i) imported fuels, especially oil, should be supplemented and replaced by local fuels, as soon as possible;
(ii) energy production from Tasmanian sources should increase to fill the supply gap created by soaring oil costs;
(iii) demand for energy must decrease;
(iv) efficiency of energy use must increase;
(v) energy losses and wastage must be reduced.

The domestic heating sector, which accounts for 12½ per cent of Tasmania's energy consumption, is too large to overlook. In terms of residential heating, the above guidelines may be interpreted to mean:

(i) oil heating should be replaced by electric, wood and coal heating (gas, as a potential fuel for transport, should not be wasted on heating);
(ii) the effects of changing heating patterns on supplies should be assessed and planned for (preferably *before* the choice of favoured heating types is made);
(iii) people should be encouraged to provide heating only when and

where it is needed;

(iv) solar heating, slow combustion heaters and heat pumps should be encouraged;

(v) incentives should be given for the use of insulation and reduction of ventilation heat losses (see Chapter Four).

The "average" Tasmanian dwelling presently uses over 28 GJ of heat per year, at a primary energy cost of 62 GJ. This is somewhat less than the 36 GJ per year (Colidcutt *et al.*, 1978) needed to provide day and evening background heat.

With full insulation, optimum siting, reduced ventilation and the use of more efficient conventional heaters, the primary energy cost of 62 GJ could be reduced to as low as 8 GJ.

Heat pumps, with seasonal COP's of two or more, could further reduce this primary energy cost to 4 GJ.

While the "4 GJ house" is unlikely to be common in the near future, this example serves to demonstrate that the potential for conserving energy in domestic heating - using known technology - is very real, and could serve to reduce Tasmania's TOTAL energy bill by as much as 10 per cent.

What is more, most of the potential for conserving heating energy has been shown to be economically favourable (Colidcutt *et al.*, 1978). The remainder of this paper describes the ways in which heat pumps can be used for domestic space heating, and evaluates the cost-effectiveness of a number of "off-the-shelf" systems.

CHAPTER TWO: DOMESTIC HEATING WITH THE HEAT PUMP

2.1 INTRODUCTION

A heat pump can heat a home at a fraction of the energy cost of a conventional heater. While the most common heat pumps to date have been of the reverse-cycle air conditioner type, there is a wide range of ways of using heat pumps for heating. It is probably true to say that the range of heat pump models is potentially as great as the range of *all* conventional heaters presently on the market.

This chapter is aimed at the potential user of a heat pump. It aims to give a *feel* for how a heat pump would fit into his home and his life-style. Section 2.2 provides a brief history of the heat pump, and explains the way in which the compression-expansion cycle moves heat from one place to another.

For practical purposes, heat pumps may be considered as three distinct parts: the heat *source* (section 2.3), the heat pump itself (section 2.5), and the heat distribution system (section 2.4). The choice of heat pump system depends on the available heat sources and the particular needs of the user. Whilst most heat pumps are electric, others (section 2.5) can use fuels ranging from gas to solar energy.

Section 2.6 explains the choice of heat pumps (electrically driven air-to-air mechanical heat pumps) for the purposes of this study, and section 2.7 compares these heat pumps with conventional heaters.

The final section summarises the findings of the chapter.

2.2 BRIEF HISTORY AND OPERATING PRINCIPLES OF THE HEAT PUMP

As early as 1852, the idea of a "heat multiplier" was proposed by Professor William Thomson, who later became Lord Kelvin (Thomson, 1852). This device could provide more heat than a furnace, from a given amount of fuel.

In an age of cheap, plentiful fuel supplies, the "heat multiplier" was little more than a curiosity. Its counterpart - refrigeration - had become well established by the end of the century. As early as 1887 it was being used to provide cool storage for Tasmanian apples (*Mercury*, April 18, 1887).

The "heat multiplier" was resurrected in Britain in about 1927, when T.G.N. Haldane built a heat pump to heat his home using heat from outside air and mains water. His machine achieved a COP of 2 to 3 (Haldane, 1930).

At about the same time Albert Einstein and Leo Szilard patented a new type of refrigerator which worked without a compressor (*New Scientist*, March 29, 1979). An ingenious combination of refrigerants needed only a heat source to set the refrigerating cycle in motion. The patent was bought by Electrolux. Zemansky (1957) describes a similar device - the Servel Electrolux refrigerator - which was invented by Carl Munters and Baltzar von Platen, in Stockholm. Such refrigerant cycles - known as *absorption* cycles - have been used in kerosene and gas/electric refrigerators. More recently, they have been used for solar air-conditioning.

After the Second World War, air-conditioners - both evaporative and refrigerated (i.e. heat pump) - became fairly common in the United States. By including a reversing valve, the refrigerated air-conditioner became a reverse-cycle air-conditioner capable of economical heating. Such machines were usually, but not always (e.g. Stoecker and Herrick, 1952) considered primarily as coolers.

Technical interest in the economy of heat pumps continued through the sixties (e.g. Mowry, 1964; Bridgers, 1967; Megley, 1968). However, high initial costs, maintenance problems and lack of understanding kept heat pumps generally out of the market.

By 1972, at least some of the manufacturers had overcome the major maintenance problems (Wilcutt, 1972). With rising oil prices, the heat pump became cost-effective in an increasing range of heating applications, and is now widely used for heating dwellings in the United States and Europe. In Tasmania, domestic heat pumps are, as yet, virtually unknown.

In commercial applications, heat pumps are well established. They provide both economical heating and cooling, at an initial cost comparable to that of the furnace and evaporative air-conditioner they replace. As mentioned in Chapter One, a number of office blocks in Hobart are heated and cooled by heat pumps.

How a Heat Pump Works

To cool the food inside it, a refrigerator must extract heat energy from it. Since energy is always conserved, the heat must be disposed of, through the condenser coil. This heat, and the heat generated directly by the motor and compressor, serve to heat the kitchen, though only slightly.

To heat a room with a heat pump requires a large heat source, and a heat pump much larger than that found in a refrigerator. One suitable source is outdoor air. By placing the evaporator coil outside, with a fan to constantly replenish the supply of air, the heat pump can, in principle, provide all the heat required.

A number of heat pump cycles are described in the ASHRAE *Handbook of Fundamentals* (1972). They include ejector, gas, thermo-electric and air. For various reasons (cost, size, inefficiency) these are rarely used. Most heat pumps, including the household refrigerator, use the *vapour compression cycle*. The bulkier and less efficient *absorption cycle* is employed in some cases (e.g. caravans) when fuels other than electricity are used.

The Mechanical (Vapour-Compression Cycle) Heat Pump

The mechanical heat pump makes use of the following two phenomena:

(i) latent heat of vaporisation (the energy required to convert a liquid to a gas at a given temperature. This energy is returned - usually as heat - when the vapour condenses back to a liquid);

(ii) the boiling point of a liquid increases with pressure; conversely, a gas can be made to condense (at a temperature above its normal boiling point) if it is compressed.

Imagine that we have some refrigerant R12 (CCl_2F_2) in a container with a movable piston for a lid (Figure 2.1). Since R12 boils at $-30^{o}C$ under normal conditions, it must be kept under pressure (360 k Pa at $5^{o}C$) to remain liquid (A).

If we now ease the pressure off the piston, the refrigerant will begin to boil (B). At first it will use its own heat to provide latent heat of vaporisation. As it becomes colder, heat will begin to flow from the ambient source to the R12 in the jar. Eventually it will absorb sufficient heat to become completely vaporised (C).

If we were to bring the container inside a house (at about $20^{o}C$), it would initially tend to absorb heat (D). However, if sufficient pressure (960 k Pa at $40^{o}C$) is applied to the piston, the vapour will be forced to condense (E). This releases the latent heat of condensation, causing the refrigerant to become hot and begin to pass heat to the house. By the time the refrigerant has completely condensed and returned to room temperature (F), each kilogram of refrigerant will have provided about 0.14 MJ of heat.

By absorbing heat energy - as latent heat of vaporisation - at a low temperature and then releasing it at a higher temperature, we have succeeded in pumping heat *up* a temperature gradient.

Figure 2.2 shows a continuous version of the cycle just described. Liquid refrigerant (A) is allowed to expand to gas in the evaporator coil (B) where it absorbs heat from the air. The vapour (C and D) is then made to condense (E_1) by the compressor and gives up its heat in the condenser coil (E_2). The cool liquid refrigerant (F) is then returned to the evaporator coil.

The Absorption Cycle

One type of absorption refrigerating cycle is described in Figure 2.3. The main point of interest is the ammonia circuit, which closely

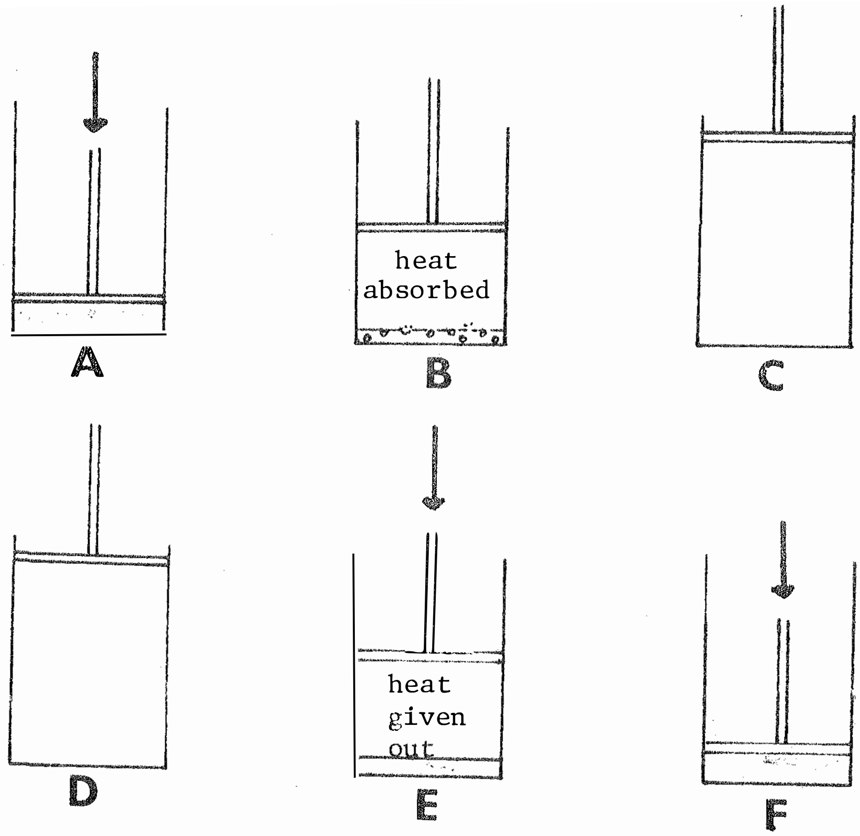


Figure 2.1: Vapour Compression cycle

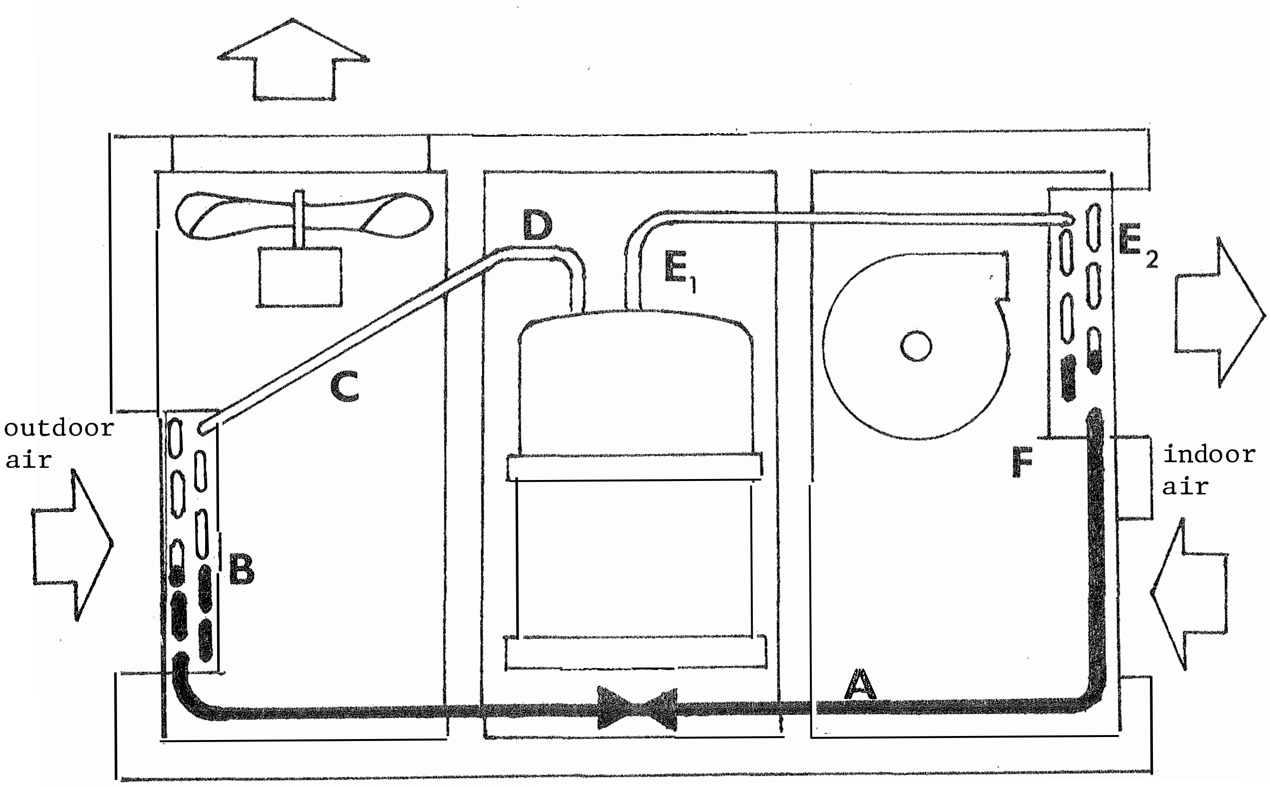


Figure 2.2: Continuous vapour compression cycle

corresponds with the refrigerant circuit in a vapour compression cycle. However, by using water to transport the dissolved refrigerant from the evaporator to the condenser, the need for a compressor is avoided.

So that the ammonia can be condensed at ambient temperatures, the whole refrigeration circuit is kept under pressure.

Other absorption cycles use a combination of water and lithium bromide. In this case, water is the refrigerant. Lithium bromide, which is extremely hygroscopic, is used (in solution) to transport the water from the evaporator to the condenser. The lithium bromide/water circuit is kept under a partial vacuum, to allow the water to be evaporated at ambient temperatures.

Efficiency

The limiting COP of a heat pump is given by thermodynamic principles as

$$\frac{273 + T_{condenser}\ (^{o}C)}{T_{condenser}\ (^{o}C) - T_{evaporator}\ (^{o}C)}$$

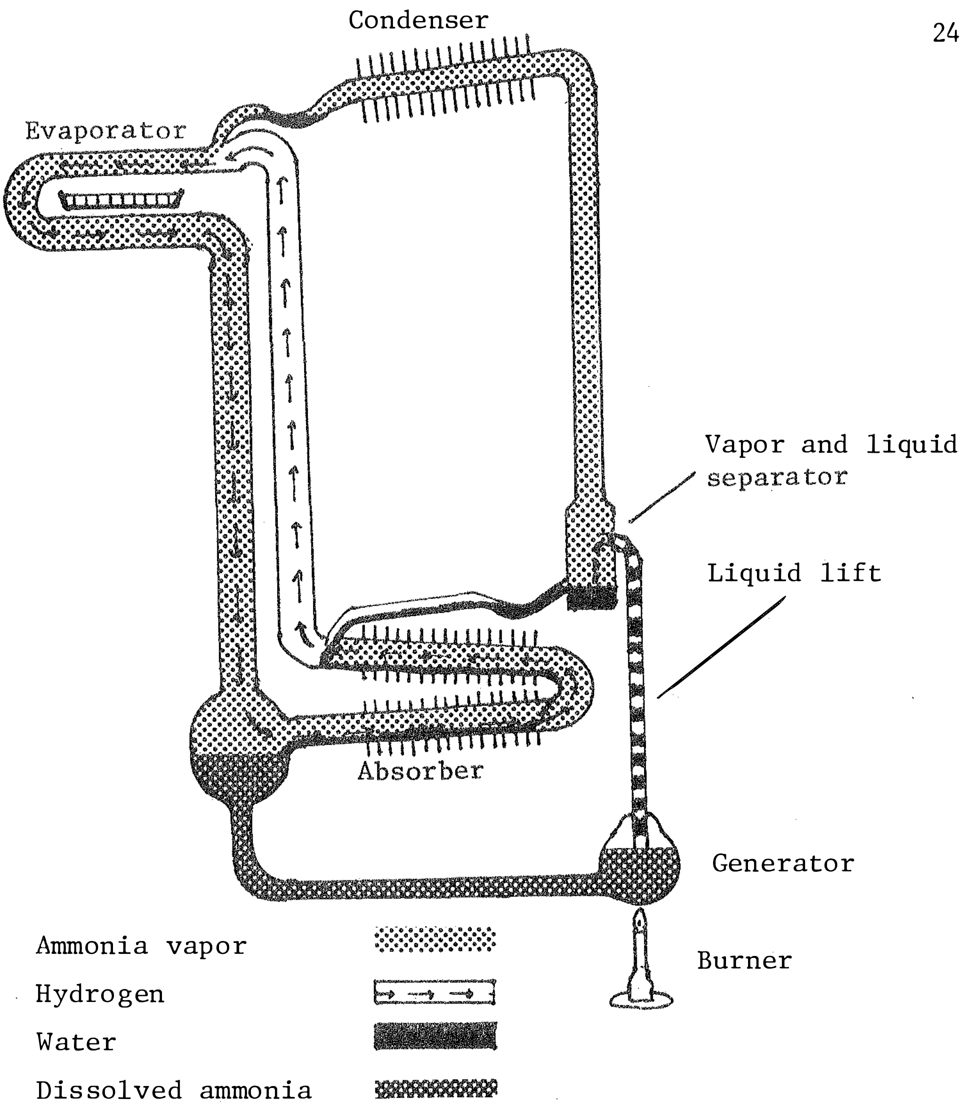
so that the heat pump becomes less efficient as the temperature gradient increases (i.e. as the temperature of the ambient source, and hence the evaporator, falls).

In practice, COP's are much lower, due to a number of factors, including the power needed to drive the fans (of an air-to-air heat pump). Vapour compression heat pumps normally have COP's of 2 to 3, while absorption heat pumps typically have *refrigerating* COP's around 0.6. Their COP's for heating - using the heat rejected in refrigeration - would be 1 $\sim$ 1.5.

2.3. HEAT SOURCES

Water

Running water is the ideal source of low-temperature heat for a heat pump. Either the evaporator is immersed directly in the water, or an antifreeze circuit is used to transfer heat from the water to the evaporator. Use of the latter method slightly lowers the COP, but is necessary if corrosion damage is likely.



The Ammonia Circuit. Droplets of water in which ammonia is dissolved separated by small amounts of ammonia vapor are raised in the liquid lift, which operates in the same way as a coffee percolator. Ammonia vapor escapes from the vapor and liquid separator, goes up to the condenser, which is cooled with the aid of radiating fins, and liquefies. It then joins a stream of hydrogen and vaporizes in an atmosphere of hydrogen while going through the evaporator, thus extracting heat from the tray of ice cubes. The mixture of ammonia vapor and hydrogen then comes in contact with water in the absorber, the ammonia dissolving and the hydrogen continuing through. The dissolved ammonia then returns to the generator.

The Water Circuit. The water that separates out in the vapor and liquid separator runs down into the absorber, where it dissolves ammonia vapor. The concentrated solution proceeds to the generator, from which it is driven into the liquid lift and raised again to the separator.

The Hydrogen Circuit. Hydrogen, which is not soluble in water, leaves the absorber and enters the evaporator, where it aids in vaporizing the liquid ammonia. The hydrogen, mixed with ammonia vapor, passes back to the absorber, where the ammonia is dissolved and the hydrogen is free once again.

(Zemansky, 1957)

Figure 2.3: Absorption heat pump cycle

Using an antifreeze circuit immersed in water at 8.3^{o}C, a COP of 2.9 can be achieved. One kilowatt of low-grade heat can be obtained by 16 metres of 19 mm dia. copper pipe. In practice, cheaper PVC piping (with similar heat transfer properties) could be used (Sumner, 1976).

Still water can also be used as a heat source. Well water, for example, can be pumped past the heat collecting coil to provide a constant supply of heat. A flow rate of 200 litres per hour will provide 1 kilowatt of low-temperature heat for a temperature drop of less than 5^{o}C.

When natural watercourses are to be used as heat sources, the minimum reliable flow rate and temperature (above 0^{o}C) must be sufficient to supply the maximum low-grade heat demand of the heat pump installed. Otherwise the water may freeze, reducing the heat output and probably causing serious damage to the compressor.

John Dawes (1978) has proposed a novel use of the heat pump for the owner of a swimming pool: the water in the pool can be used as a heat source. A large domestic swimming pool, containing 75,000 litres of water, can provide 1½ GJ (420 kWh) of low-grade heat for a 5^{o}C drop in temperature. With 200 m^2 of surface area, it could recover its lost heat, by conduction and convection, at a rate of 3½ kW, or up to 90 kWh per day. This would be sufficient to heat the living zone of an insulated house (see Chapter Five). In summer, with the heat pump cycle reversed, the heat extracted by air-conditioning the house would provide "free" heating for the swimming pool.

Unfortunately, water in the quantities needed for heating is not generally available.

Earth

Heat can be obtained from the ground by a heat collecting coil buried at a depth of 1 to 2 metres. To save on expensive refrigerant, and minimise maintenance, the buried coils usually use an antifreeze solution to transfer heat from the earth to the evaporator.

The size of the excavation and length of coil required depend on the thermal properties of the soil, the annual heat load and the peak heating power. One excavation in England provided 0.6 GJ per m^2, over a year. Nineteen millimetre copper tubes collected 14 ∿ 17 watts of heat per lineal metre, at COP's of 2.3 ∿ 2.5. Black plastic hose, laid on the ground, collected 10.4 watts per lineal metre (Sumner, 1976).

These results suggest that the 20 GJ, and 6 kW, of heating required to heat the living zone of an insulated house (see Chapter Five) could be provided by 600 metres of plastic hose laid evenly in an excavation of 32 m^2, at a cost of around $320 for the excavation and $375 for the plastic tubing. However, the thermal parameters of soils vary widely, by a factor of two or more. On-site testing of the soil would be required before the performance of an actual earth-source heat pump could be predicted.

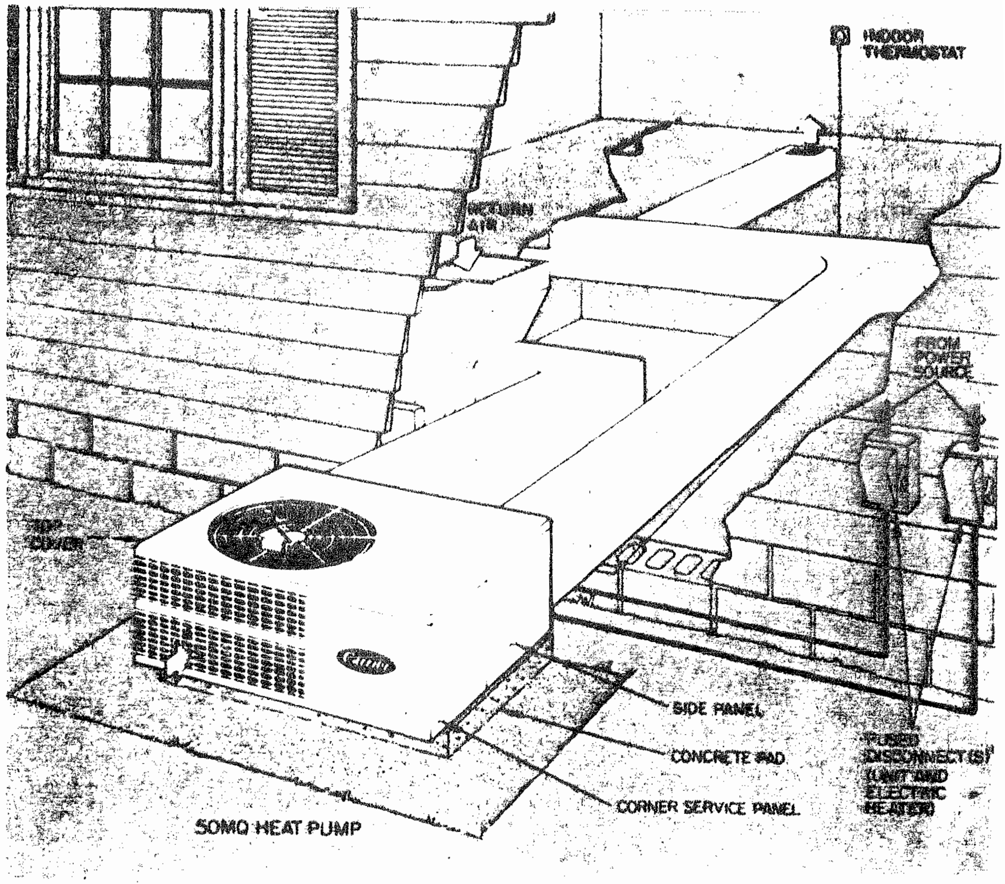
The excavation cost could be avoided by laying the coil with the foundations of the house. A single length of plastic hose, 50 metres, would provide less than one kilowatt of low-grade heat, and would need to be supplemented by another heat source.

Air

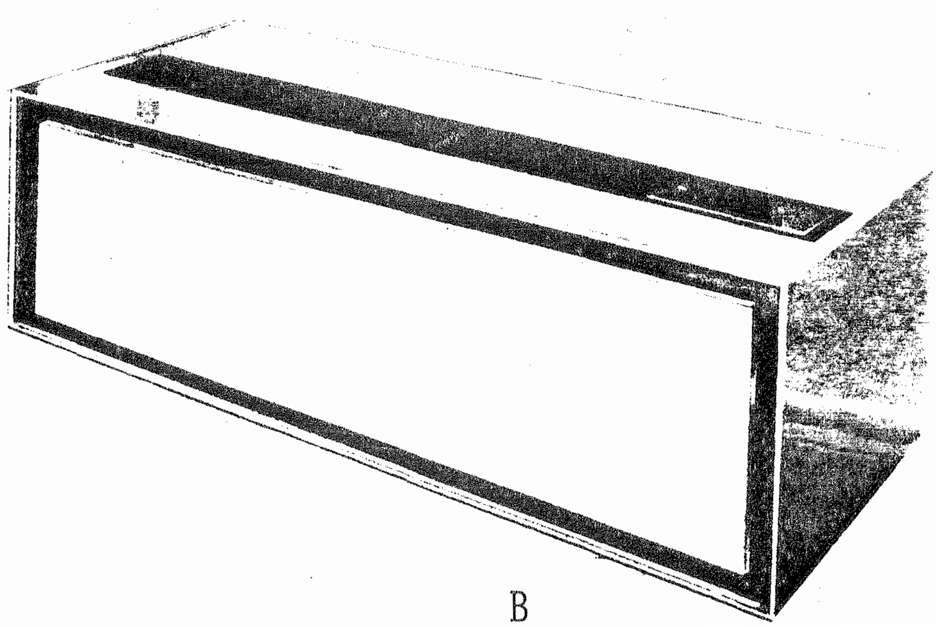
Because it is universally available, air is the most common source of heat for heat pumps. Air-source heat pumps have grown out of the air-conditioning industry, and are available as single package units (Figure 2.4) or split systems (Figure 2.5).

The only specific requirement of the air-source heat pump is that the evaporator must be outside of the heated space. Single package units are either mounted through an external wall, or located completely outside the house, supplying warm air to the house via ducting. Split systems have an outdoor section containing the evaporator and compressor, and an indoor section containing the condenser and air-handling equipment. The two sections are connected by up to ten metres of refrigerant piping, which is considerably cheaper than an equivalent length of two-way ducting. However, the split system must be charged with refrigerant, by a mechanic, after installation.

There are two major disadvantages of using air as a heat source. The first is that the COP of the heat pump falls as the temperature drops. At the lowest temperatures, when most heat is needed, the air source heat pump is least able to provide the required heat. The second is



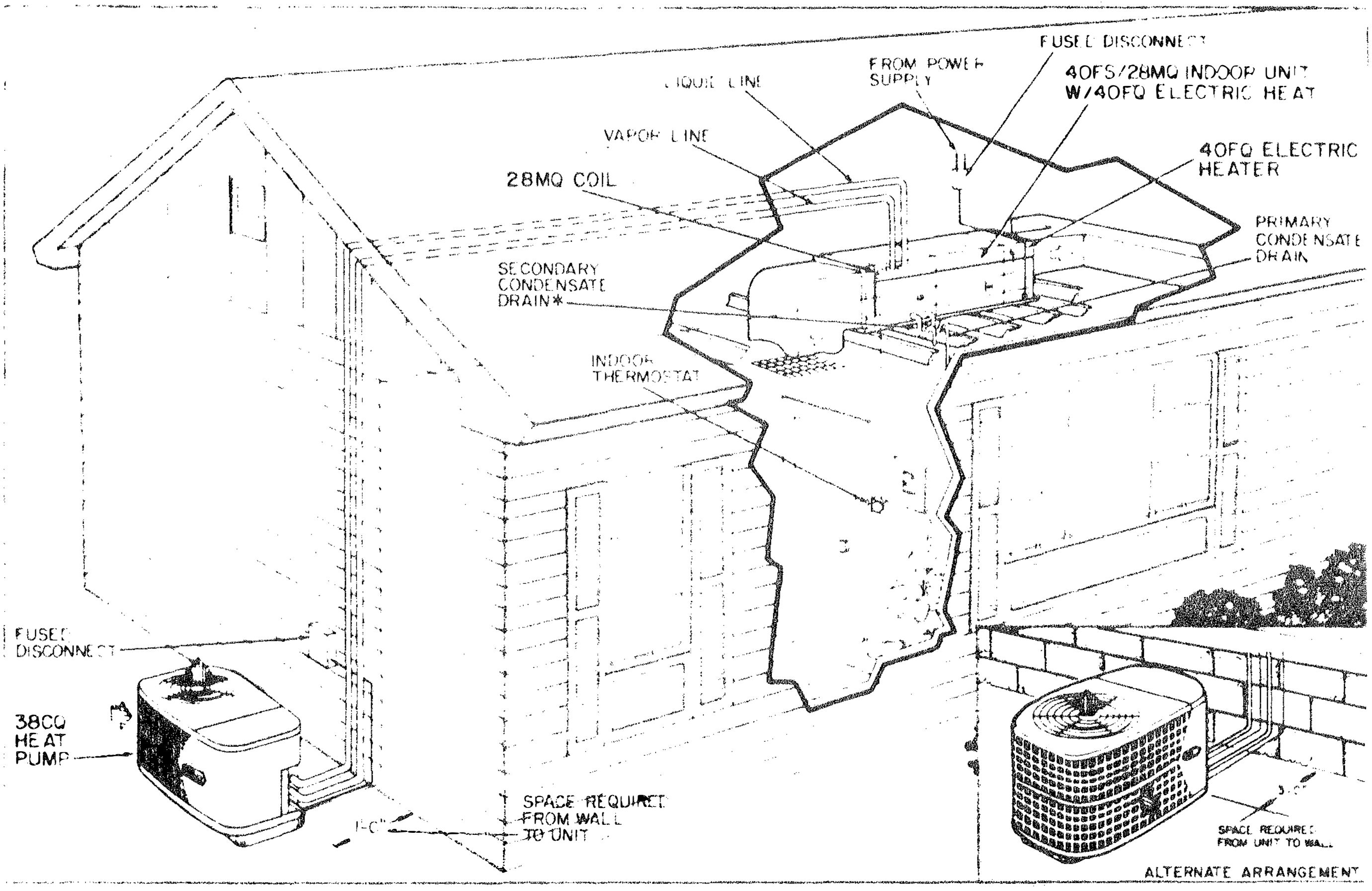
A



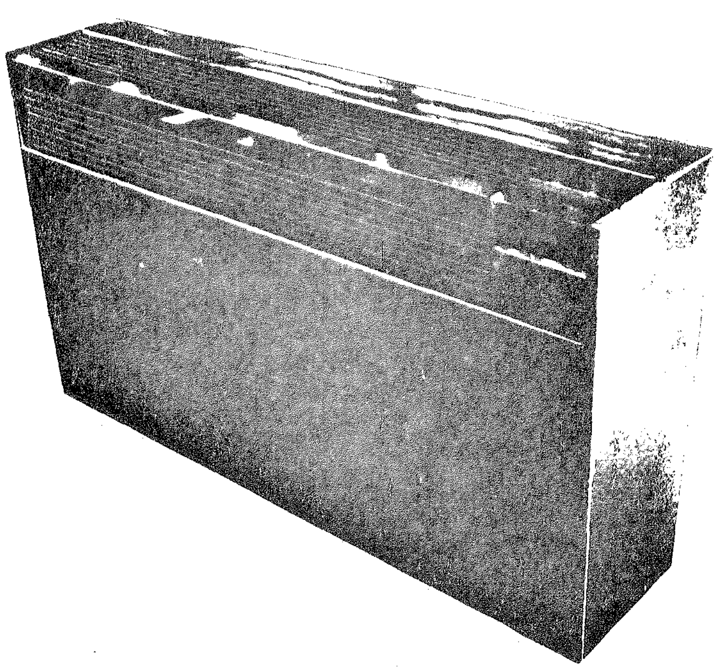
B

Figure 2.4: Single package heat pumps: A: Ducted system

B: Through-the-wall unit.



A



B

Figure 2.5: Split system heat pumps:

A: Ducted system with indoor air handling unit.

B: Indoor console unit.

that at temperatures below 6°C, frost forms on the evaporator coil, reducing its effectiveness. The frost is removed either by reversing the cycle, or by small heaters placed near the coil. Neither method is entirely satisfactory - both reduce the COP, and reversing the cycle lowers the net heat output. A further drawback of reversing the cycle is that it increases reliability problems.

One side-effect of the air-source heat pump, which is sometimes useful, is its dehumidifying effect. Moisture from the air condenses onto the evaporator coil and can be collected by a container under the coil. If the evaporator is placed where moisture is a problem but cold is not - e.g. a damp cellar or ceiling - the moisture problem can be controlled.

The category of air-source heat pumps includes most reverse-cycle air-conditioners. When both cooling and heating are required, such machines have a unique ability to fill both roles. In Tasmanian homes, however, they have a number of disadvantages over heat pumps designed for heating:

(i) there is little need for their cooling ability (see Chapter Three);

(ii) they are designed primarily for cooling hot air down to around 25°C; for the best heating performance (up to 20°C) the design criteria (e.g. type of refrigerant; compressor ratio) are different;

(iii) reverse-cycle air-conditioners without defrosting facilities cannot be used below about 6°C;

(iv) cool air inlets are best mounted high in a room, so that the cool air falls and spreads through the room; warm air outlets, on the other hand, are best mounted low.

Solar

Solar-assisted heat pumps offer advantages over either pure solar or pure heat pump heating systems. Because the heat pump can use solar heat at relatively low temperatures, cheaper solar panels, with less insulation, can be used. Because solar panels provide heat at higher temperatures than other sources, they allow the heat pump to operate at a higher COP. Thus, by combining solar collectors and heat pumps, a given amount of heat can be provided by cheaper solar collectors and a smaller heat pump operating more efficiently.

To make proper use of these advantages, the system should be designed as a whole, rather than using components designed for other purposes.

Smetana (1976) has achieved COP's of 3.5 using solar-assisted heat pumps, and Andrews *et al.* (1978) suggest that COP's from 3 (at 5^{o}C evaporator temperature) to 5 (30^{o}C) or more are feasible. Since commercial solar collectors can reliably generate temperatures up to 50^{o}C in Hobart, the efficiency of a solar-assisted heat pump is potentially quite high.

Solar energy can also be used to directly power a heat pump, operating on an absorption cycle. A solar air-conditioner using the principle is described in the CSIRO, Division of Mechanical Engineering, Information Service Bulletin No. 12/A/6 (April, 1978). A 7-8 kW cooling system is estimated to cost around $9000 plus installation charges.

There is nothing to prevent such a system being used for heating, by reversing the flow of cooling water. The system would have a heating capacity of around 15 kW under the same conditions (but less in winter).

A simple means of taking advantage of solar energy would be to place the evaporator between the ceiling and roof of a house. To obtain the greatest benefit, the ceiling should be insulated and the roof should be a colour that will absorb radiant heat. During the day, the air under the roof is warmed by the sun, permitting the heat pump to obtain a higher COP.

The problem of intermittent operation of solar heating devices can be overcome either by using some form of heat storage, or by using a heat pump which can obtain heat from ambient sources when solar heat is not available.

Stored Heat

Heat can be stored in water, in solids (e.g. rock piles), in the structure of the house itself (see Chapter Four) or as latent heat in a phase change of a suitable material.

By using stored heat, the heat pump can achieve a higher COP and avoid the need for defrost cycles.

Andrews *et al.* (1978) describe a solar heat pump system which successfully pumped heat into ground storage early in the heating season to allow it to meet peak loads later in the season. Heat can be supplied to storage by heat pumps or solar-assisted heat pumps (when they are not needed for house heating) or direct from solar collectors.

Perhaps the best known form of storage is the rock pile or gravel bed, heated by warm air. A large tank of water can also be used as a heat store.

The phase change of water to ice has been used (*New Scientist*, January 12, 1978, p. 90) for heat storage. The latent heat of fusion of water (Figure 1.1) is quite considerable, and can be obtained as heat by re-freezing the water. Furthermore, such a heat store needs no insulation.

Because heat pumps cannot efficiently reach high temperatures, high temperature heat storage (as used in off-peak electric storage heaters) is not feasible. One phase change of Glauber's salt occurs at 32.4°C, well within the range of heat pump temperatures, and suitable for home heating. At one time, canisters of Glauber's salt, in canvas bags, were used as foot-warmers in Victoria's trains. However, the salt has a tendency to degrade after only a few heating/cooling cycles.

To store 50 kWh as latent heat of fusion - comparable to the heat storage of a large electric heat bank - would require 750 kilograms of Glauber's salt, occupying 0.5 m^3 and costing around $900.

Waste Heat

In industry, where "waste" heat may be produced in large quantities, heat recovery by heat exchangers, heat pumps, or heat wheels (Department of Energy, 1977) can provide economical heating for various purposes. In supermarkets, the refrigeration equipment pumps its heat directly into the aisles, making considerable savings on winter heating bills.

In homes, heat is wasted in the forms of hot water (down the drain) and warm air (through ventilation heat losses). Ventilation accounts for about one-quarter of the house's total heat loss.

Neither of these heat sources could supply all the heating requirements of a house, and the expense of a specially designed heat recovery system does not appear to be warranted. However, if a heat pump is used for heating, either or both of these sources could be used to boost the COP.

For example, using an air-source heat pump, natural ventilation could be replaced by a forced ventilation system feeding directly onto the evaporator coil. This input of warm ($20^{o}C$) air would increase the COP and virtually eliminate the need for de-icing.

2.4 HEAT DISTRIBUTION

Most air-conditioners and heat pumps use forced (or ducted) air to distribute the heat within a house. Water - pumped through pipes in a concrete floor, or through hot-water radiators - can also be used. Radiant heating, which requires very high temperatures, cannot be achieved using heat pumps.

Air

A forced convection heater can warm up the inside of a house in about thirty minutes, compared with several hours for any heater which must first heat the structure of the house. The only faster method of heating is by using radiant heat. So for intermittent heating with a heat pump, air is the preferred method.

Water

For continuous heating, water may be the preferred medium of heat transfer. In some cases, water pipes may prove cheaper to install than bulky air ducts. In some existing houses it will be impossible to fit air ducting under the floor, and water may be the only suitable means of distribution.

With heat pumps, it is usually necessary to use a water temperature of $60^{o}C$, rather than the $80^{o}C$ design temperature for hot water radiators. To compensate for this, they need to be increased in size by about one-third.

Much better heat distribution is obtained if a concrete slab floor is used and the slab itself is heated. This is currently done only with electric in-floor heating. To heat the slab with hot water pipes would require approximately 70 metres of pipe per kilowatt of heating. Plastic pipe can be used, at around $44 for 70 metres.

A major advantage of using water as a heat distributor is that it can easily be turned on or off in various parts of the house, so that heat is not wasted by being used where it is not wanted. A further advantage is that it is virtually silent, needing no fans.

The prime advantage of air is that it can heat up an area more rapidly than hot water radiators or in-floor heating.

2.5 HEAT PUMP MECHANISMS

Electric Heat Pumps

In Tasmania, electricity is the obvious choice for driving a heat pump. Hydro-electric generation is rated 85 per cent efficient. So an electrically driven mechanical heat pump (COP 2.5) has an EER of 2.1, compared with 0.85 for resistance heating.

Engine Driven Heat Pumps

A mechanical heat pump can also be driven by a fuel-fired engine. With an efficient engine, such a machine can heat more efficiently than a furnace. The search for an efficient engine has led to a revived interest in external combustion engines and, particularly, in the Stirling engine.

A modern Philips Stirling engine is 25.4 per cent efficient at 3000 rpm (Wurm, 1975).

Coupled to a heat pump, the Philips Stirling engine would achieve an EER of 0.63 - somewhat less than a furnace! If the heat given off by the engine could be used effectively for heating, the EER would rise to around 1.1.

The main problem with the Stirling engine is that a large (i.e. expensive) engine is required for a modest power output. If a 40 per cent efficient Stirling engine could be economically produced and coupled to a heat pump, an EER of around 1.3 would be obtained.

Martini (1977) has designed a combined Stirling engine - heat pump, which, he says, can approach within 80 per cent of the limiting Carnot efficiency.

A solar powered turbocompressor heat pump has been tested by Biancardi and Meader (1975). Solar heat is used to boil the refrigerant. The refrigerant vapour drives a turbine, in the same way that steam drives a conventional turbine. The turbine in turn drives a compressor, which operates a vapour compression cycle. In this design the power cycle and the heat pump cycle have been combined, and a COP of 0.45 (cooling) was achieved. Reclaiming the solar heat could bring the heating up to 1.45. The authors predict that a cooling COP of 1 can be achieved. This would mean a heating COP of up to 2.

In Britain, the Department of Energy is spending over $2 million on heat pump research. Much of this will be spent on heat pumps driven by gas-fired engines. One company is developing a 10 kW domestic model, which it wants to sell at around $1300 as a substitute for a boiler. It is expected to save $200 a year on fuel costs compared with a $300 boiler, and to have a payback time of five years (*New Scientist*, May 31, 1979, p. 727).

Engine-driven heat pumps remain very much in the area of research. A prototype 18 kW heat pump to be installed at the Open University during 1979 will be the first gas heat pump to heat a building in Britain (*New Scientist*, May 31, 1979, p. 727).

Absorption Heat Pumps

A less complicated way of using low-grade energy sources (wood, coal, gas, etc.) to run heat pumps is by use of the absorption cycle. With efficient heat recovery, absorption heat pumps could be twice as fuel-efficient as conventional furnaces.

Using solar power, an absorption heat pump could increase the effective output of solar panels by supplementing the solar heat with heat from a low-temperature source. Because absorption heat pumps require heat at 45°C or above, reasonably well insulated panels would be needed in Tasmania. Such devices would only compete with direct solar heating if the absorption units cost less than the 35-40 per cent saving on collector area.

It appears that the only current use of absorption heat pumps in Tasmania is in small mobile refrigerators (for caravans, etc.). Elsewhere, they have been used for cooling, in larger capacities. Apparently, they are too expensive in smaller sizes (below 10 kW) to compete with mechanical refrigeration.

Perhaps because the absorption machines are not easily reversed, they have not until now been considered in a heating role.

At present, the absorption heat pump may be the most fuel-efficient means of heating, using low-grade energy sources. However, it is likely that they will soon be overtaken in this regard by engine-driven heat pumps.

2.6 HEAT PUMPS FOR TASMANIAN CONDITIONS

Many of the heat sources and heat pumps described in sections 2.3 and 2.5 have attractive prospects. However, for a study such as this, which aims to determine the feasibility of heat pumps for broad application in the immediate future, the costs and performance of the heat pump must be accurately known. The following types of heat pumps have been chosen for evaluation:

Heat source:	air
Heat distribution:	air
Heat pump type:	electrically driven vapour compression heat pump, single unit or split system.

Most domestic heating in Tasmania is presently done by heaters with outputs in the range 2 kW (portable electric heaters) to 11 kW (oil heaters) with larger capacity units for central heating. Heat pumps with rated outputs from 2 kW to 10 kW will be considered in this report.

In smaller heating capacities, the most readily available form of heat pump is the reverse-cycle air-conditioner. Having been on the market for several decades, its costs and performance are well known. As air-conditioners are designed primarily for cooling, the heating COP can vary quite significantly. One well-known company (Email) even markets a model which heats by resistance rather than reverse cycle. Not all air-conditioners have de-icing facilities, which are necessary for safe low-temperature operation, and few (particularly in smaller capacities) offer the option of supplementary heating.

The Building Research Establishment in Britain has designed and built a 2 kW heat pump expressly for heating (Freund and Cattell, 1979). However, a 2-6 kW air-conditioner, with a higher rated COP at a comparable cost, was chosen in preference.

Smaller single-unit air-conditioners, which can be installed through walls and windows and run from household power outlets, are extremely cheap to install.

Heat pumps in the range 5-8 kW can be used to heat the whole living area of a house. In open-plan areas, the cost of ducting can be avoided by using a through-the-wall unit, or a split system with an indoor heating console. If a ducting is used, the same heat pump can be used to heat the sleeping area as well, in a *changeover* system. This system divides the house into two zones. The living zone is heated during the day and evening, and the sleeping zone at night. Discomfort in the unheated zone is reduced by the effects of thermal storage and heat leakage from the heated zone (see Chapters Four and Five).

For continuous heating to the whole house, a larger ducted heat pump (9 kW +) will be required. With these systems, the whole heat pump (single package) or the compressor and outdoor coil (split system) are located away from the house, by up to ten metres.

Specifications of a number of heat pumps suitable for heating living areas are shown in Table 2.1. Ratings for the air-conditioners are only given at standard conditions (7°C dry bulb outdoor temperature). For these, it is assumed that heat output is constant above 6°C, and that the compressor operates at maximum power through the range*. Below 6°C, when defrosting will be required, outputs are assumed to fall to 1.5 kW, 2 kW and 4 kW for the respective units. Performance ratings for the 5.3 kW and 6.8 kW units are shown in Figures 2.6 and 2.7. In comparison with these, it is likely that the overall performance of the

* Output and COP normally increase with outdoor temperature, but the amount of increase is not known for the reverse-cycle air-conditioners.

Table 2.1: Specifications of heat pump models chosen for evaluation.

Make/Model	Nominal output[1] kW	Nominal input[1] kW	Nominal COP[1]	Basic price[2] $	Installed cost[3] $	Maintenance[4] $ per year	Running cost[5] $ per 20 GJ
HITACHI RA-104 CHA (r/c air-conditioner)	2.6	1.08	2.44	561	750	30	76
HITACHI RA-144 CHA (r/c air-conditioner)	3.8	1.7	2.235	757	950	45	83
HITACHI RA-2255 CHA-1 (r/c air-conditioner)	6.4	3.15	2.05	1049	1300	60	90
G.E. Weathertron WA918 (split system, ducted, 3 outlets)	5.3	2.6	2.03		1600-2000	60	91
POPE Super Eagle (single package, ducted, 4 outlets)	6.8	2.72	2.50	1002	2000-2800	70	74
POPE Console CQ21[6] (split system)	6.7			1467	2000	70	

[1] Standard conditions for rating heat pumps are 7°C outdoor temperature (dry bulb), 21°C indoor temperature. Nominal input includes compressor input plus fan power, and for the air-conditioners is the maximum input power.

[2] Basic price for the air-conditioners is recommended retail price (Hobart). Australian price for the General Electric model is not available. Prices for the POPE models are ex-factory (Melbourne) including sales tax.

[3] Installed cost, for the Hitachi models, includes basic price, installation, thermostat and electrical wiring. For the ducted systems, the minimum of $900 has been allowed for ducting, including ducting from the heat pump to the indoor plenum (also included).

[4] Maintenance includes routine maintenance plus an annual allowance of 5 per cent of the cost of compressor replacement.

[5] Running cost is calculated at the nominal COP. For comparison with conventional heaters, see Table 2.2.

[6] The console model can be expected to perform as well as the Super Eagle, as it requires slightly lower fan power (hence has a slightly improved COP). It is included to show the relative costs of ducted and console systems.

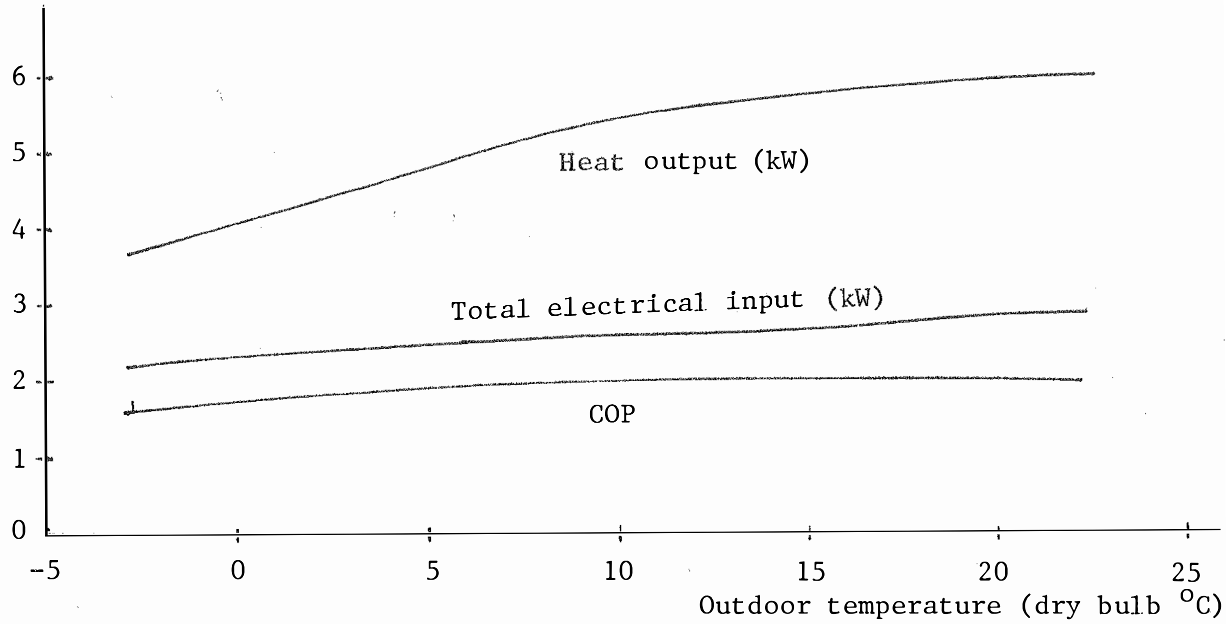


Figure 2.6: Performance specifications of 5.3 kW heat pump ($21^{o}C$ indoor temperature).

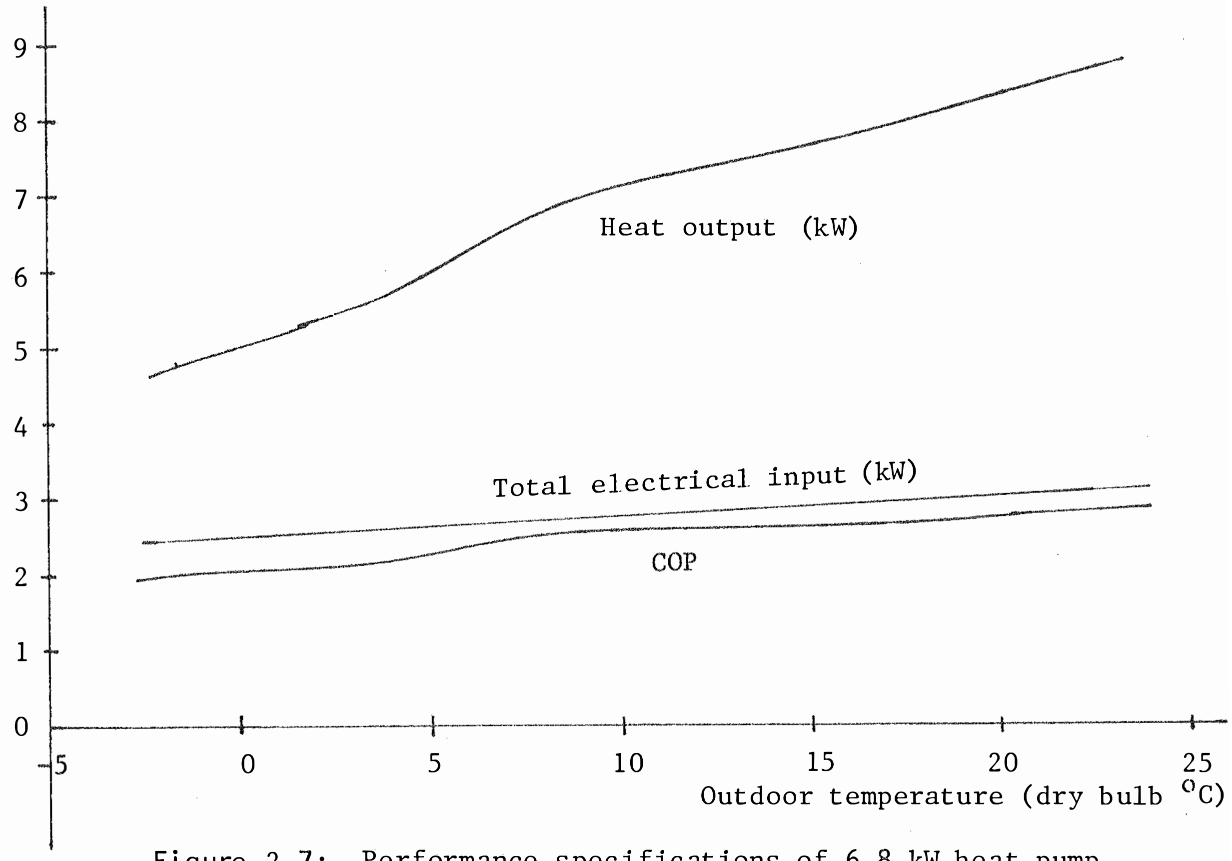


Figure 2.7: Performance specifications of 6.8 kW heat pump ($21^{o}C$ indoor temperature).

air-conditioners has been underestimated throughout the range.

Maintenance costs shown include routine maintenance, plus an allowance of 5 per cent of the cost of replacing a compressor, in case of breakdown.

Since the air-conditioners have no supplementary heating fitted, auxiliary heating will be needed to provide adequate heating at low temperatures. Normally, this would be electric heating operating on the household tariff.

2.7 COMPARISON OF HEAT PUMPS WITH CONVENTIONAL HEATERS

Economic Factors

Heat pumps (Table 2.1) have higher initial and maintenance costs than conventional heaters (Table 2.2). This is because they are relatively complex, and have more precise installation requirements. Running costs can be estimated by dividing the cost of resistance heating ($185 per year) by the *seasonal* COP of the heat pump installation. Since heat pumps are generally designed to meet substantially less than 100 per cent of the peak load, a proportion of the heating will be done by auxiliary resistance heating, reducing the COP below the nominal value. Seasonal COP's range between 2.5 and 1.5, with corresponding running costs of $75 to $125, depending on the particular heat pump installation.

Heaters can be grouped according to their running costs:

- Low: slow combustion heaters, electric heat banks, some heat pumps.
- Medium: small-bore electric, electric midi- and combi-banks, oil heaters, other heat pumps.
- High: electric (household tariff), gas, open fires.

Due to their high installed costs, heat pumps (and small-bore electric hot water systems) do not appear to compete with electric heat banks or slow combustion heaters. Oil heaters are expected to become uneconomic within five years, and the 40 per cent of homes currently (A.B.S., 1978) using oil heaters will be switching to off-peak (currently 7 per cent), slow combustion (12 per cent) or more expensive forms of heating.

Because of their high installed costs and low running costs, heat

Table 2.2: Summary of heating costs, conventional heaters.

Heater type	Rated output[1] kW	Installed cost[2] $	Maintenance $ per year	Running cost[3] $ per 20 GJ
Oil	10.8	500	20	134
Gas	5	450	15	209
Wood: open fire	?	800	8	200
slow combustion	13	400	10	61
Electric: direct		70 per kW	5	185
off-peak midi bank	3.5	400	5	78
off-peak heat bank	6	750	5	78
off-peak floor heating	10	1200	5	78
small-bore	? (whole-house heating)	4500	10	117

1 For off-peak electric heating, the figure given refers to the rated *input*; rated outputs of other heaters are corrected for efficiency (see Appendix A).

2 Installed cost allows for purchase, installation, gas cylinders, etc. The cost of an open fire is derived from the basic cost of a fireplace, as calculated by the Tasmanian Housing Division ($740). Other costs have been determined from a survey of retail outlets in Hobart, during June of 1979.

3 Running cost is calculated from efficiency (Appendix A) and energy contents and prices of fuels (Appendix B). 20 GJ can provide day and evening background heating for the living area of an insulated (ceiling only) house, for one heating season.

pumps are most economic when a large total amount of heating is required from a given unit. If they are to be used for a short period each day, they are unlikely to be competitive. If the heating period extends through the day, then the increased energy savings, in many cases, can make them economic.

Heat pumps are also favoured in buildings which are of heavy construction. A heavy building (e.g. cavity brick walls, concrete slab floors) needs about the same *total* amount of heating as a light building (e.g. weatherboard), but it will keep a more even temperature. The light building will be colder at night and warmer during the day. As a result, it will need a heater of higher capacity than the heavy building needs. Thus, it will not be as economic for heat pump use.

A more complete economic comparison is made in Chapter Six.

Comfort and Convenience

For convenience, direct electric heaters (including heat pumps) are superior to all others. The more sophisticated electric heaters offer thermostatic control and time-switches.

Gas heaters also are easy to start, and have potential for thermostatic control. Oil heaters require several minutes to warm up, and have a limited range of heat settings. In mild conditions, an oil heater even on its lowest setting can overheat a room. Extended running on low heat settings also causes soot buildup.

Off-peak heaters usually need direct electric heating as a back-up for the coldest periods. The cheaper off-peak heaters cannot be turned on and off at will. If heating is needed for less than ten hours per day, direct electric heaters can be cheaper to run. This problem is largely overcome by the larger heat banks, which use a fan to bring out their heat and otherwise store it until it is required.

Wood heaters lose out when it comes to convenience. Aside from the need to chop wood, there is a warm-up time of around half an hour before the fireplace or heater becomes hot enough to be an effective heater. Temperature control with wood heating requires repeated attention to fuel levels.

On comfort criteria, heat pumps are superior to most popular heaters.

Michell and Biggs (1978) have shown (Figures 2.7 to 2.10) that oil heaters, gas space heaters, electric wall furnaces and heat banks produce poor heat distribution patterns. Floor-heated, ducted and forced-air heaters (Figures 2.11 to 2.13) distribute the heat well.

Poor heat distribution can result in stuffiness (when the head is $3^{o}C$ warmer than the feet), cold feet and/or high fuel bills (when the heat goes mostly to the ceiling). Additionally, it encourages draughts. Thus, a heater should firstly produce enough heat, and secondly distribute it evenly around the room.

For convection heaters a good heat distribution is provided by a relatively large volume of warm (not hot) air directed horizontally near floor level. The first two of these criteria are, by coincidence, natural requirements of heat pumps (which do not produce very high temperatures). Provided the heat pump outlet is near floor level, good heat distribution will be achieved.

The other heat distribution methods used with heat pumps - ducted air and in-floor water heating - also distribute the heat evenly.

The main disadvantage of the heat pump in this regard is that it is not a very good "personal" heater. Radiant heaters (electric, oil, gas or wood) remain the best for thawing out cold bodies.

Noise

The fans that supply air to the indoor and outdoor coils of an air-to-air heat pump generate a certain amount of noise. Noise is also produced by the compressor.

By careful design and the use of soundproofing materials, heat pumps have been made considerably quieter than "old-fashioned" air-conditioners. Split-system heat pumps allow the major causes of noise - the compressor and outdoor fan - to be placed outside the house. With ducted systems, the complete heat pump package can be placed outside, circulating warm air or water through the house for heating.

Most heat pumps produce an audible background noise in the region of 50 dB to 70 dB. Outdoor units need to be reasonably quiet, and to be placed where the noise they produce will not disturb neighbours.

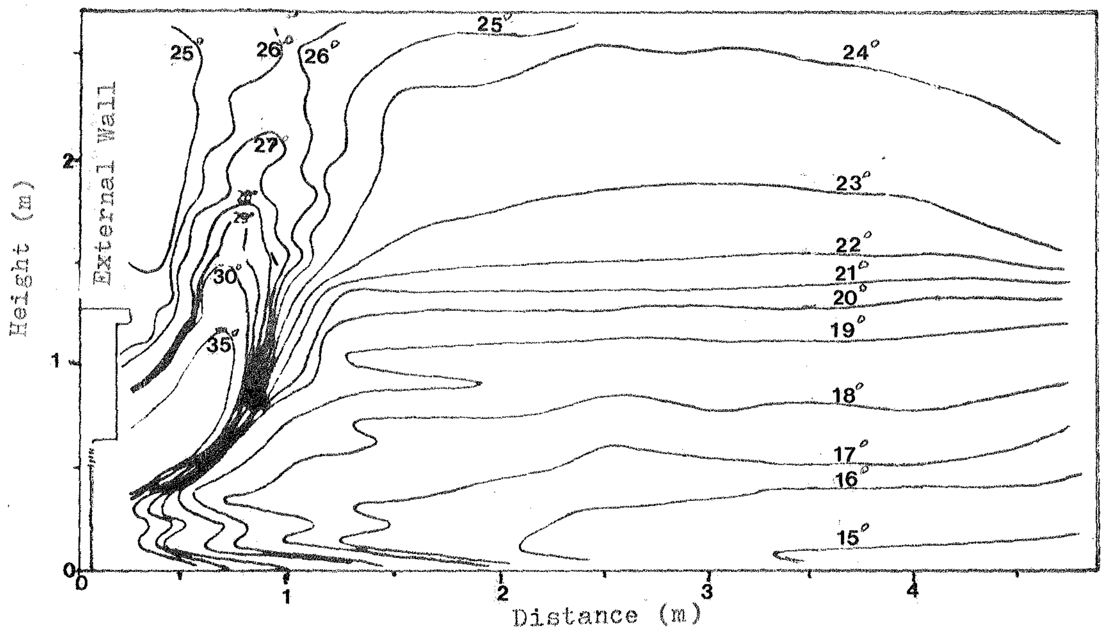


Figure 2.7: Distribution of temperatures with oil space heater. Isotherms at 1 and 5°C intervals. Outside temp. 6°C. Outlet air temp. ≈ 70°C. Outlet air velocity ≈ 1.25 m/s. (Michell and Biggs, 1978)

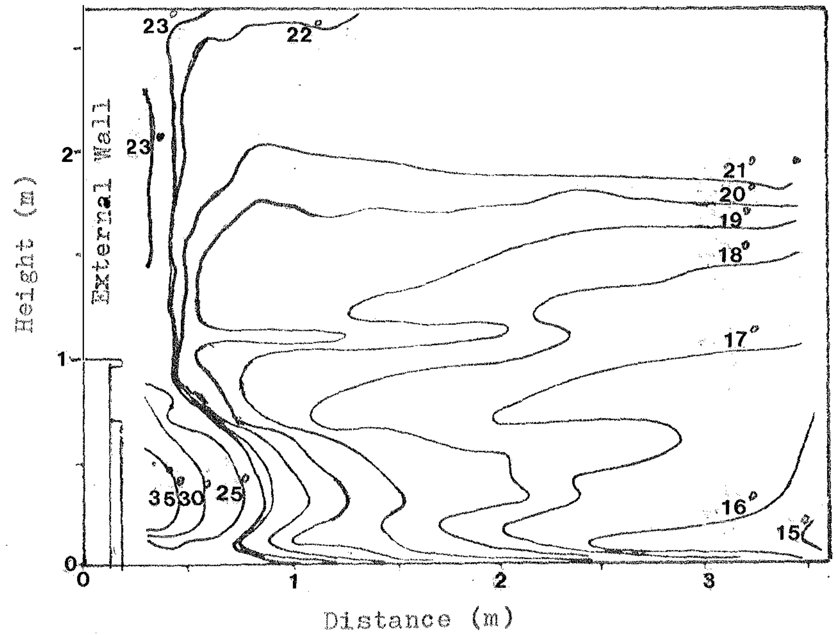


Figure 2.8: Distribution of temperatures with gas space heater. Isotherms at 1 and 5°C intervals. Outside temp. 14°C. Outlet air temp. ≈ 105°C. Outlet air velocity ≈ 0.75 m/s. (Michell and Biggs, 1978)

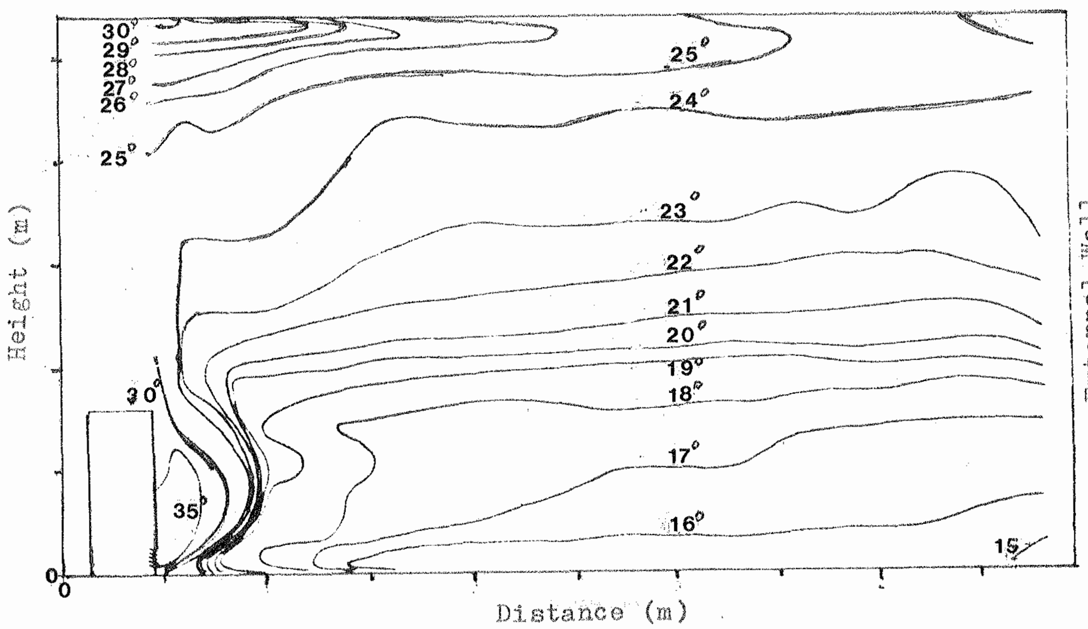


Figure 2.9: Distribution of temperatures with electric heat bank.

Isotherms at 1 and 5°C intervals. Outside temp. 14°C.

Outlet air temp. ≈ 80°C. Outlet air velocity ≈ 1.33 m/s.

(Michell and Biggs, 1978)

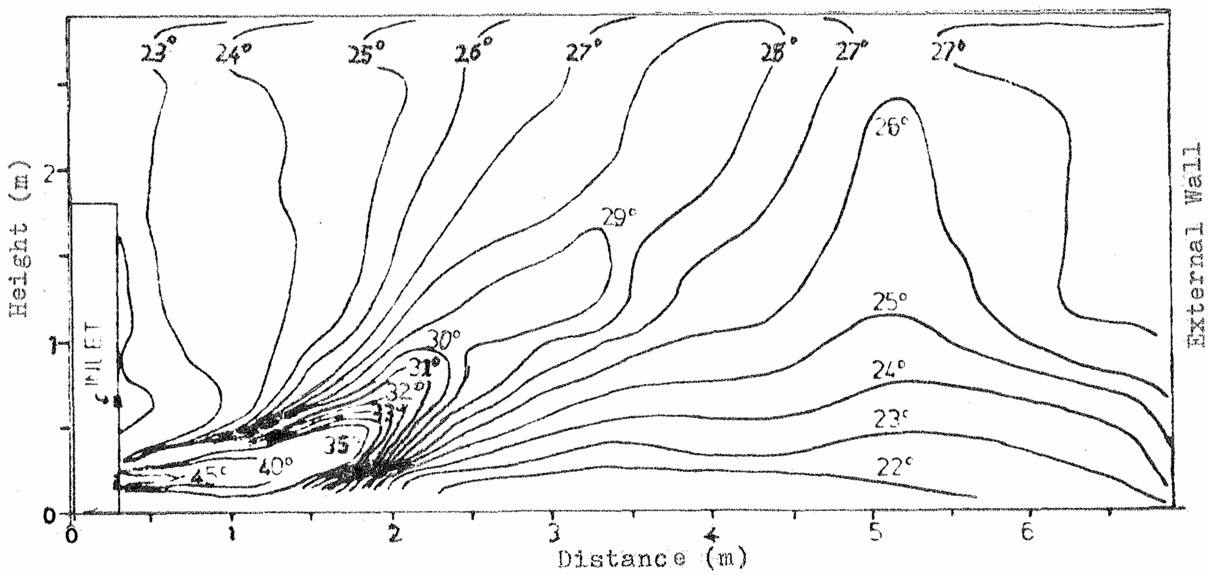


Figure 2.10: Distribution of temperatures with electric wall furnace.

Isotherms at 1 and 5°C intervals. Outside temp. 8°C.

Outlet air temp. ≈ 74°C.

Outlet air velocity ≈ 2.3 m/s.

(Michell and Biggs, 1978)

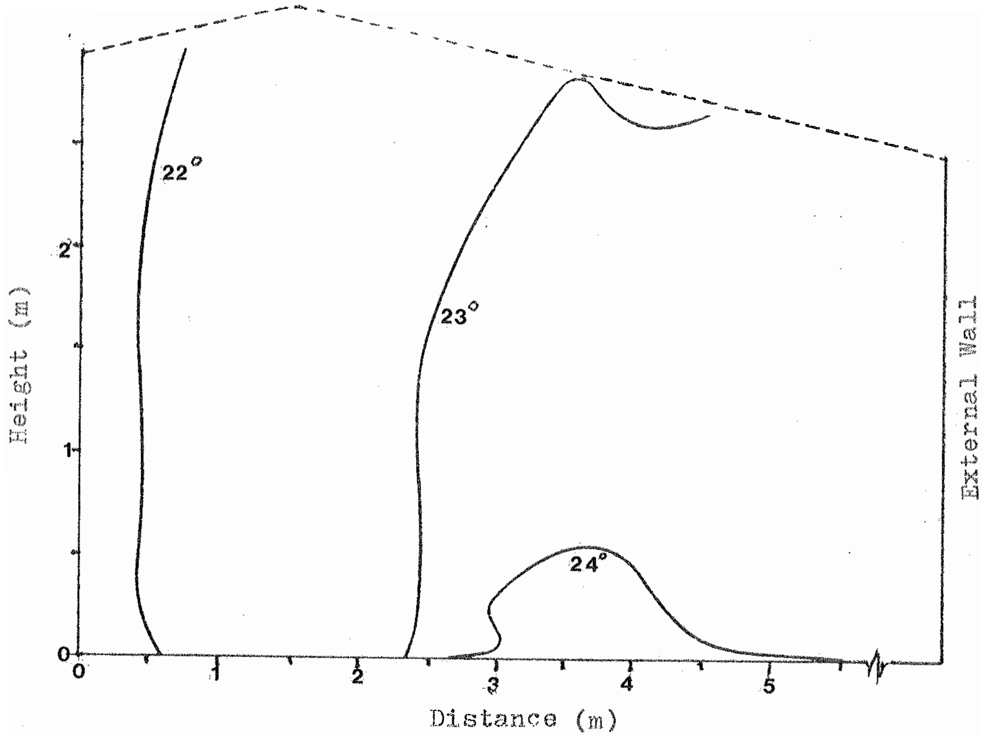


Figure 2.11: Distribution of temperatures with water-heated concrete floor. Isotherms at 1°C intervals. Outside temp. 12°C.

(Michell and Biggs, 1978)

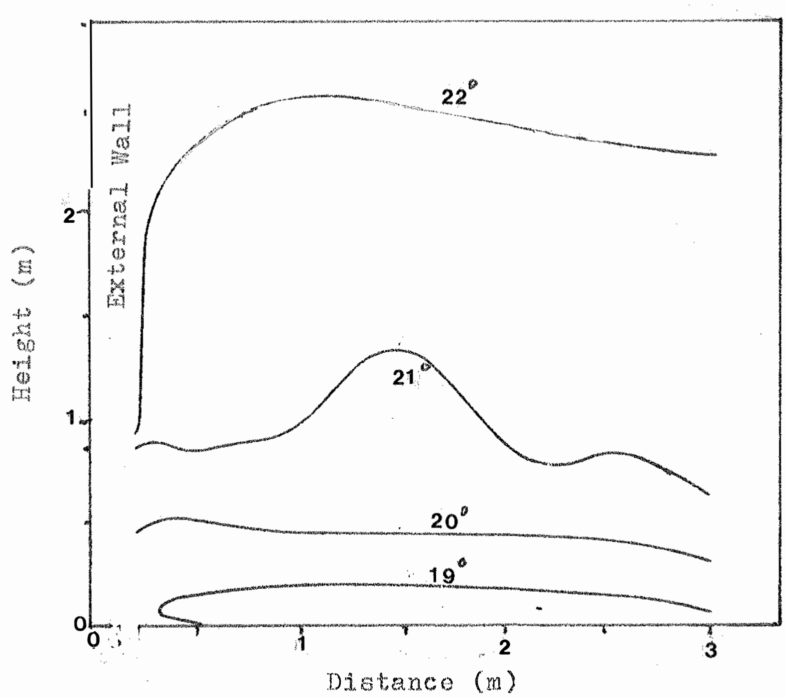


Figure 2.12: Distribution of temperatures with ducted heating system. Isotherms at 1°C intervals. Outside temp. 11°C. Outlet air temp. = 44°C. Outlet air velocity = 2.2 m/s.

(Michell and Biggs, 1978)

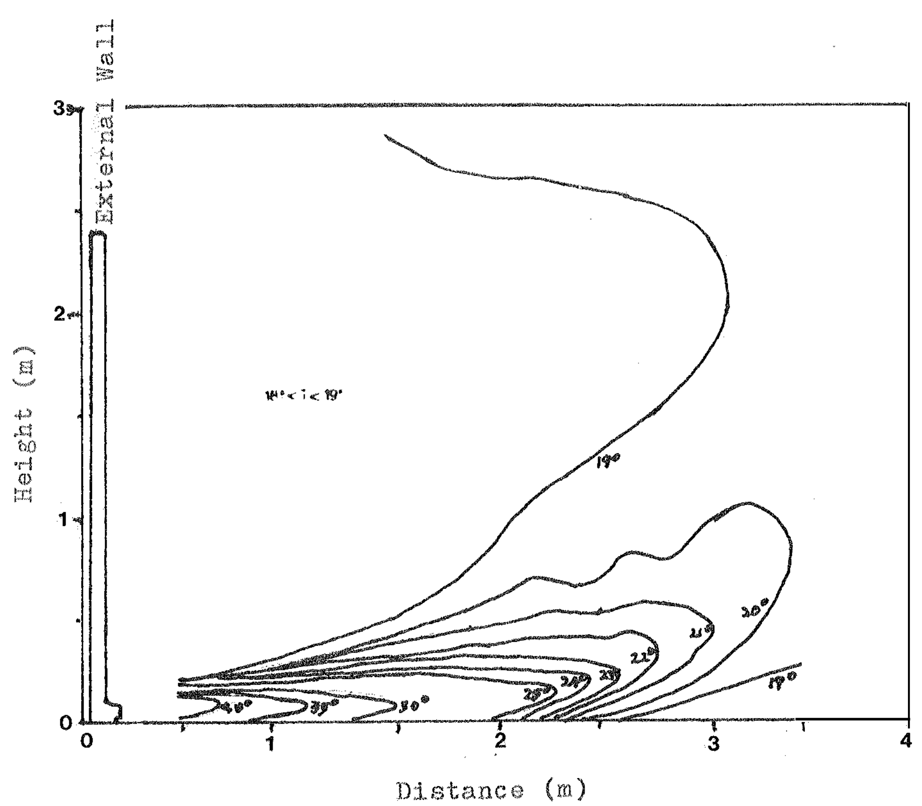


Figure 2.13: Distribution of temperatures with forced-air heater Isotherms at 1°C and 5°C intervals. Outside temp. 12°C. Outlet air temp. ≃ 55°C. Outlet air velocity ≃ 3.33 m/s.

(Michell and Biggs, 1978)

Maintenance and Lifetime

The most expensive type of breakdown in a heat pump is a compressor failure. Replacing a compressor costs several hundred dollars, depending on the size of the compressor, and it can be a major part of the total servicing cost.

A comprehensive study of heat pump reliability (Wilcutt, 1972) analysed the service history of almost five thousand heat pumps over an average of 2.1 years. Nine major brands were divided into three groups according to reliability.

On average, the heat pumps had an annual service cost of U.S.$33.73, and a compressor failure rate of 6.13 per cent per annum. The average interval between service calls (excluding installation, customer education and incomplete or non-essential calls) was thirteen months.

Considerable variation was found between brands. The best group averaged U.S.$21.40 annual service cost, a 3.22 per cent compressor failure rate and sixteen months between calls. At the other end of the scale, 26 of the original 150 models were taken off the Approved Heat Pump list in 1972, because of unreliability.

With continuing development of heat pumps, a general standard of reliability comparable to that of the best group of manufacturers in 1972 could be assumed for the 1980's.

Wilcutt (1972) has also pointed out that heat pumps need a high standard of installation. The electrical wiring system may be a problem to the average heating and cooling contractor, and poorly designed or badly fitted ducting will prevent the unit from performing to expectations.

In practice (Akalin, 1978) heat pumps have an average lifetime of eleven years. Associated hardware, such as ducting, is expected to last from twenty years to the lifetime of the house, making replacement of a ducted heat pump considerably cheaper than the initial installation. Because of the possibility of compressor failure, maintenance contracts would be an advantage. A system of five-year maintenance contracts, as used by the Alabama Power Company (Wilcutt, 1972), could also ensure a high quality of installation and identify less reliable models.

2.8 SUMMARY AND CONCLUSIONS

By obtaining "free" heat from a low temperature source, a heat pump can provide heating, using less of our energy resources than conventional heaters.

Of the available ambient heat sources, probably the best is running surface water. Heat can be distributed by ducted air or pumped hot water. The heat pump can be driven by an electric motor, a small engine, or - in the case of absorption heat pumps - a simple heat source.

For general applications in Tasmania in the near future, the most likely prospect is an air-to-air electric heat pump. Air-to-water heat pumps can also be considered for central heating systems. Noise levels and reliability of modern heat pumps are acceptable, but service contracts should be available. These heat pumps have high initial costs compared with conventional heaters, but their day-to-day running costs are among the lowest. They are expected to be economic when heating is required for the major portion of the day.

Heat pumps produce a better *quality* of heat than conventional heaters, are cheap to run and in certain cases are expected to be more economical overall than most conventional heaters.

CHAPTER THREE: HEATING FOR A COMFORTABLE ENVIRONMENT

3.1 INTRODUCTION

Because heat pumps have relatively high fixed costs and low running costs, their economic viability is ultimately determined by the total amount of heat required.

When a building is heated constantly to a particular temperature, its heat loss exactly balances the heat input from the heater. So a good estimate of the heat loss is also a good estimate of the heat required to maintain the building at that temperature. The heat loss can be accurately calculated from three sets of information:

(i) indoor temperature(s)
(ii) thermal properties of the building
(iii) climate of the locality of the building.

The calculated heat loss is approximately proportional to the difference between indoor and outdoor temperatures. When dealing with an air-to-air heat pump this temperature difference becomes doubly important, since it also determines the heat pump's COP.

Comfort standards (including temperatures) for indoor heating are available. These standards usually refer to the conditions that should be provided in office buildings. At home, the heating may be regulated according to different criteria. The conditions for comfort are discussed in section 3.2, and a comfort "standard" for Tasmanian households is derived in section 3.3.

Section 3.4 explains how Hobart's climate has been "predicted" for the purpose of calculating heating requirements. It also answers some questions on the reliability of such predictions, and the application of these calculations to other locations in Tasmania.

Thermal properties of buildings will be considered in Chapter Four.

3.2 THERMAL COMFORT

Thermal comfort may be defined as "that condition of mind which expresses satisfaction with the thermal environment" (Shaw and Stephenson, 1977). A person experiences thermal comfort when he or she is affected by a suitable combination of thermal influences. The main factors affecting thermal comfort are depicted in Figure 3.1.

The human body constantly produces heat. To be in thermal comfort, it must lose just the right amount of heat to its environment - too much, and the body becomes cold; too little, and it becomes hot. The body can maintain thermal *equilibrium* over a wide range of conditions by shivering (cold) or sweating (hot). The range of conditions for comfort (as opposed to equilibrium) is much narrower.

Because it is difficult to mathematically treat all the factors that affect thermal equilibrium, and because comfort requirements vary from person to person, thermal comfort is best treated as a *subjective* phenomenon. In evaluations of subjective comfort, a seven-point scale is commonly used:

Subject's Rating	Assigned Numerical Value (vote)	Comfort Rating
hot	+3	warm discomfort
warm	+2	
slightly warm	+1	comfort
neutral	0	
slightly cool	-1	
cool	-2	cool discomfort
cold	-3	

The subject ratings and numerical values shown are as used by Fanger (1970). Individual responses may differ, but the numerical mean of a large number of subjects' ratings will always approach the same value (for a given set of conditions).

From a large amount of experimental data, Fanger has derived a "comfort equation". This equation uses the thermal variables to

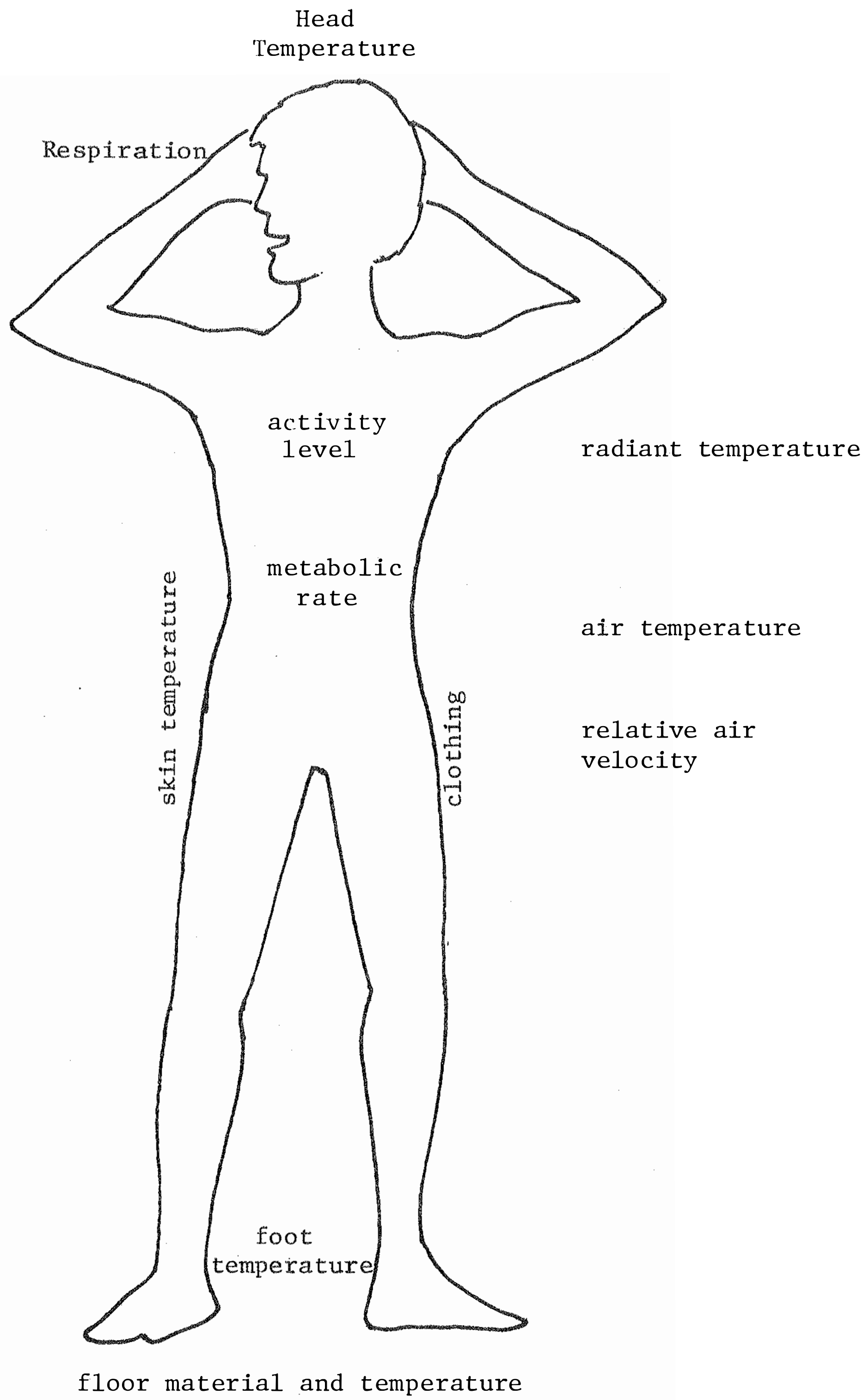


Figure 3.1: Major factors affecting thermal comfort

predict the mean numerical rating expected from a large group of subjects. Optimum conditions for thermal comfort are achieved by adjusting the variables to produce a "predicted mean vote" (PMV) of zero.

Fanger's comfort equation is quite complex, and is not easily solved. To facilitate its use, Fanger has calculated values of PMV for a wide range of conditions.* PMV is presented as a function of activity level, clothing, ambient temperature and relative air velocity (these terms will be defined later in this section). The effects of less important factors, such as humidity, can be adequately estimated by corrections to the values given by Fanger.

Even under optimum conditions (PMV = 0) not all subjects will be at thermal comfort. In fact, no standard conditions can be expected to provide comfort for more than 95 per cent of subjects. More important is the percentage of dissatisfied people under a given set of conditions. As conditions change from optimum, the total number of dissatisfied people in a group will increase. The proportion of dissatisfied people can be related to the mean vote of the group. Likewise, the "predicted proportion dissatisfied" (PPD) is related to the PMV, as shown in Figure 3.2. Since the thermal variables cannot be maintained *exactly* at any optimum combination, it is important to know how far they can be allowed to change without causing a large increase in the PPD.

For practical purposes, factors affecting thermal comfort can be classified as either *personal* (e.g. metabolic rate, clothing) or *environmental* (e.g. temperature, humidity). Usually, a person will choose his own activities and clothing. The normal function of a heater is to alter one of the environmental variables (temperature) to make the person comfortable under the given conditions.

A practical heating standard can be derived by specifying all the thermal parameters except temperature. The optimum temperature corresponds to PMV = 0 for the given conditions.

For an activity level of 80 k cal. m^{-2} hr^{-1}, a clo-value of 1.0, a relative air velocity of 0.1 $m.s^{-1}$ and 50 per cent relative humidity, the optimum temperature is found to be 18.2^{o}C. These conditions will be used as a starting point for evaluating the influences of the individual variables, in the discussion below.

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* See Table 3.3 for a selection of PMV values.

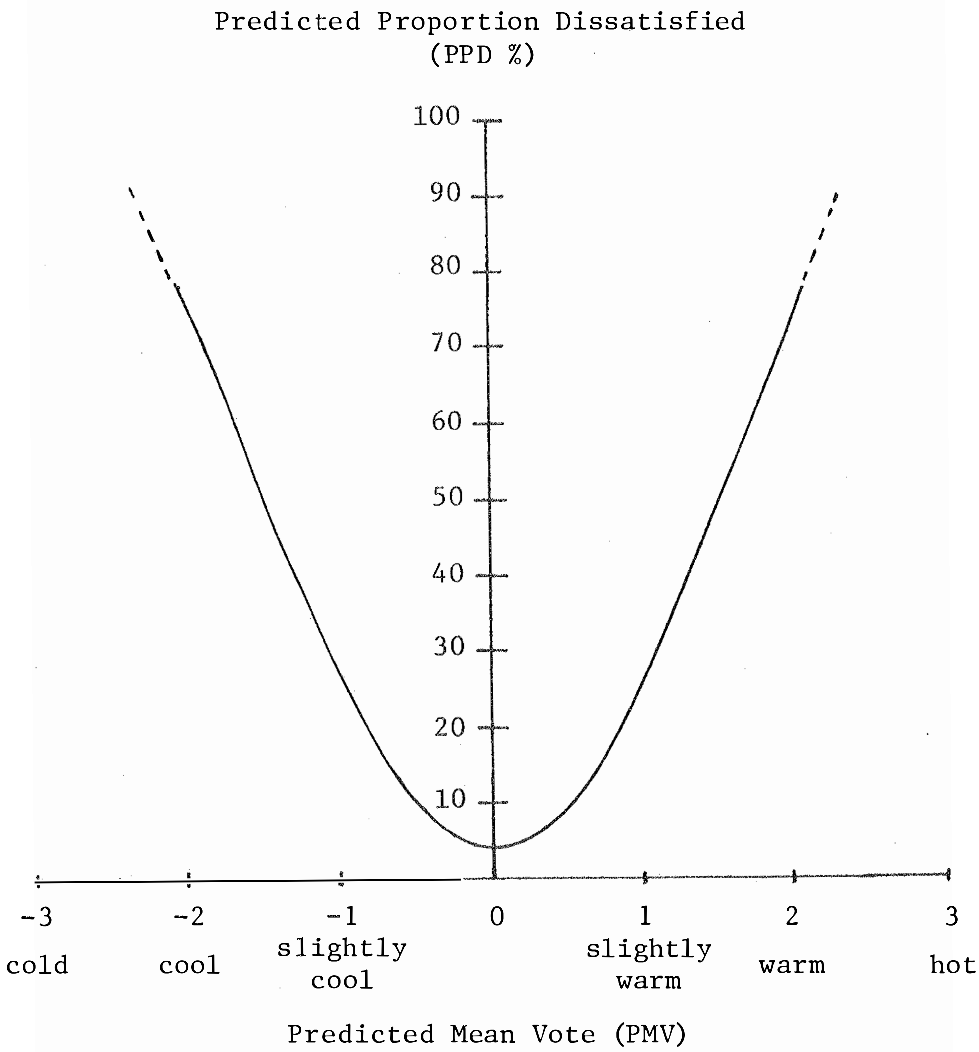


Figure 3.2: Relationship between comfort (PMV) and discomfort (PPD)

Activity Level

The activity level is the amount of heat generated by metabolic processes. Consider the human body as a machine, converting metabolic energy into mechanical work at an efficiency η. As in a machine, the power input is converted either into work or heat. The power input is the metabolic rate (M). Thus, the body produces work at the rate $M\eta$, and heat at the rate $M(1-\eta)$. This heat is passed to the surroundings through the body surface. For convenience, the heat production is expressed as heat output per unit of body surface area (du Bois area A_{Du}). When it is expressed this way, it becomes independent of body size. The heat production per unit of body size, for a given activity, is also relatively independent of body size. Metabolic rates and efficiencies can be physically determined for most activities. Some examples are shown in Table 3.1.

As might be expected, activity level has a major influence on thermal comfort. In Figure 3.3, the effects on PMV and PPD of varying the activity level are shown, in comparison with the effects of varying the other thermal parameters. The variations relate to the standard conditions mentioned above.

Reducing the activity level from 80 k cal. m^{-2} hr^{-1} to about 50 kcal. m^{-2} hr^{-1} changes the thermal sensation from neutral to slightly cool. Increasing it to 130 k cal. m^{-2} hr^{-1} results in a vote of slightly warm.

Metabolic heat also contributes to the heating of a building. In calculations of the air-conditioning load of office buildings, it is customary to add allowances for metabolic heat and the heat produced by lights and other appliances. The contribution of metabolic heat can be estimated from activity levels. A sedentary person has a metabolic rate of 50 k cal. m^{-2} hr^{-1}. The average person has a surface area of around 1.77 m^2. Hence, the heating contribution of a sedentary person is 88.5 k cal. hr^{-1}, or 103 watts. Moderate housework doubles this contribution to around 200 watts.

Clothing

Clothing provides thermal resistance to the body's loss of metabolic heat. Heavier clothing reduces the body's loss of heat, causing the

Table 3.1: Metabolic rates for selected activities.[1]

Activity	Metabolic rate M/A_{Du} $kcal.hr^{-1}.m^{-2}$	Mechanical efficiency η	Relative velocity in still air $m.s^{-1}$
Sleeping	35	0	0
Seated, quiet	50	0	0
Standing, relaxed	60	0	0
Walking (level, 4.8 km/h)	130	0	1.3
House cleaning	100 ~ 170	0 ~ 0.1	0.1 ~ 0.3
Cooking	80 ~ 100	0	
Washing dishes, standing	80	0	0 ~ 0.2
Shaving, washing and dressing	85	0	0 ~ 0.2

[1] Compiled from a table in Fanger (1970).

Table 3.2: Data for different clothing ensembles (from Fanger, 1970).

Clothing Ensemble	I_{cl} (clo)
Nude	0
Typical tropical clothing ensemble	0.3-0.4
Light summer clothing	0.5
Typical business suit	1.0
Heavy traditional European business suit	1.5

body to become warmer. Thus, PMV is higher when heavier clothing is worn.

The thermal resistance of clothing is measured by a term called the "clo-value", I_{cl}. The clo-value is a dimensionless term for the total thermal resistance from the skin to the outer surface of the clothed body. Examples of clo-values are given in Table 3.2.

It can be seen in Figure 3.3 that clothing has an important role in determining comfort. This fact can be used to advantage by the 5 per cent of people who experience either cool or warm discomfort under optimum standard conditions. They can compensate for their discomfort by wearing more or less clothes. At home, where they can choose their own personal optimum conditions, they may compensate by varying the temperature rather than their clothing.

Relative Air Velocity

A large part of the body's heat loss occurs by convection, from the outer surface of the clothing. Some heat loss also occurs by evaporation of sweat, though under comfort conditions the rate of sweat secretion is not usually large. Both of these heat losses increase as wind speed increases. The more vigorous activities (Table 3.1) generate their own relative air velocities, even in still air. For the low relative velocities experienced in buildings, the effect on comfort is moderate (Figure 3.3).

Humidity

As well as by evaporation of sweat, the body loses latent heat in respiration. Air is breathed in at normal atmospheric temperature and humidity. It is breathed out at approximately 34^{o}C and 3 per cent (weight for weight) absolute humidity ratio. The latent respiration heat loss depends on the amount of water vapour added to the air by the lungs. So the latent respiration heat loss will be lower at higher humidities, as will be the latent heat loss from sweating.

As shown in Figure 3.3, humidity has only a small effect on thermal comfort.

Table 3.3: Predicted mean vote (selected values from Fanger, 1970).

Activity Level 50 kcal/m^2hr

Clothing clo	Ambient Temp. °C	Relative Velocity (m/s) <0.10	0.10	0.15	0.20	0.30	0.40	0.50	1.00	1.50
0	26.	-1.62	-1.62	-1.96	-2.34					
	27.	-1.00	-1.00	-1.36	-1.69					
	28.	-0.39	-0.42	-0.76	-1.05					
	29.	0.21	0.13	-0.15	-0.39					
	30.	0.80	0.68	0.45	0.26					
	31.	1.39	1.25	1.08	0.94					
	32.	1.96	1.83	1.71	1.61					
	33.	2.50	2.41	2.34	2.29					
0.25	24.	-1.52	-1.52	-1.80	-2.06	-2.47				
	25.	-1.05	-1.05	-1.33	-1.57	-1.94	-2.24	-2.48		
	26.	-0.58	-0.61	-0.87	-1.08	-1.41	-1.67	-1.89	-2.66	
	27.	-0.12	-0.17	-0.40	-0.58	-0.87	-1.10	-1.29	-1.97	-2.41
	28.	0.34	0.27	0.07	-0.09	-0.34	-0.53	-0.70	-1.28	-1.66
	29.	0.80	0.71	0.54	0.41	0.20	0.04	-0.10	-0.58	-0.90
	30.	1.25	1.15	1.02	0.91	0.74	0.61	0.50	0.11	-0.14
	31.	1.71	1.61	1.51	1.43	1.30	1.20	1.12	0.83	0.63
0.50	23.	-1.10	-1.10	-1.33	-1.51	-1.78	-1.99	-2.16		
	24.	-0.72	-0.74	-0.95	-1.11	-1.36	-1.55	-1.70	-2.22	
	25.	-0.34	-0.38	-0.56	-0.71	-0.94	-1.11	-1.25	-1.71	-1.99
	26.	0.04	-0.01	-0.18	-0.31	-0.51	-0.66	-0.79	-1.19	-1.44
	27.	0.42	0.35	0.20	0.09	-0.08	-0.22	-0.33	-0.68	-0.90
	28.	0.80	0.72	0.59	0.49	0.34	0.23	0.14	-0.17	-0.36
	29.	1.17	1.08	0.98	0.90	0.77	0.68	0.60	0.34	0.19
	30.	1.54	1.45	1.37	1.30	1.20	1.13	1.06	0.86	0.73
0.75	21.	-1.11	-1.11	-1.30	-1.44	-1.66	-1.82	-1.95	-2.36	-2.60
	22.	-0.79	-0.81	-0.98	-1.11	-1.31	-1.46	-1.58	-1.95	-2.17
	23.	-0.47	-0.50	-0.66	-0.78	-0.96	-1.09	-1.20	-1.55	-1.75
	24.	-0.15	-0.19	-0.33	-0.44	-0.61	-0.73	-0.83	-1.14	-1.33
	25.	0.17	0.12	-0.01	-0.11	-0.26	-0.37	-0.46	-0.74	-0.90
	26.	0.49	0.43	0.31	0.23	0.09	0.00	-0.08	-0.33	-0.48
	27.	0.81	0.74	0.64	0.56	0.45	0.36	0.29	0.08	-0.05
	28.	1.12	1.05	0.96	0.90	0.80	0.73	0.67	0.48	0.37
1.00	20.	-0.85	-0.87	-1.02	-1.13	-1.29	-1.41	-1.51	-1.81	-1.98
	21.	-0.57	-0.60	-0.74	-0.84	-0.99	-1.11	-1.19	-1.47	-1.63
	22.	-0.30	-0.33	-0.46	-0.55	-0.69	-0.80	-0.88	-1.13	-1.28
	23.	-0.02	-0.07	-0.18	-0.27	-0.39	-0.49	-0.56	-0.79	-0.93
	24.	0.26	0.20	0.10	0.02	-0.09	-0.18	-0.25	-0.46	-0.58
	25.	0.53	0.48	0.38	0.31	0.21	0.13	0.07	-0.12	-0.23
	26.	0.81	0.75	0.66	0.60	0.51	0.44	0.39	0.22	0.13
	27.	1.08	1.02	0.95	0.89	0.81	0.75	0.71	0.56	0.48
1.25	16.	-1.37	-1.37	-1.51	-1.62	-1.78	-1.89	-1.98	-2.26	-2.41
	18.	-0.89	-0.91	-1.04	-1.14	-1.28	-1.38	-1.46	-1.70	-1.84
	20.	-0.42	-0.46	-0.57	-0.65	-0.77	-0.86	-0.93	-1.14	-1.26
	22.	0.07	0.02	-0.07	-0.14	-0.25	-0.32	-0.38	-0.56	-0.66
	24.	0.56	0.50	0.43	0.37	0.28	0.22	0.17	0.02	-0.06
	26.	1.04	0.99	0.93	0.88	0.81	0.76	0.72	0.61	0.54
	28.	1.53	1.48	1.43	1.40	1.34	1.31	1.28	1.19	1.14
	30.	2.01	1.97	1.93	1.91	1.88	1.85	1.83	1.77	1.74
1.50	14.	-1.36	-1.36	-1.49	-1.58	-1.72	-1.82	-1.89	-2.12	-2.25
	16.	-0.94	-0.95	-1.07	-1.15	-1.27	-1.36	-1.43	-1.63	-1.75
	18.	-0.52	-0.54	-0.64	-0.72	-0.82	-0.90	-0.96	-1.14	-1.24
	20.	-0.09	-0.13	-0.22	-0.28	-0.37	-0.44	-0.49	-0.65	-0.74
	22.	0.35	0.30	0.23	0.18	0.10	0.04	0.00	-0.14	-0.21
	24.	0.79	0.74	0.68	0.63	0.57	0.52	0.49	0.37	0.31
	26.	1.23	1.18	1.13	1.09	1.04	1.01	0.98	0.89	0.84
	28.	1.67	1.62	1.58	1.56	1.52	1.49	1.47	1.40	1.37

Table 3.3: (Continued)

Activity Level 80 kcal/m²hr

Clothing clo	Ambient Temp. °C	Relative Velocity (m/s) <0.10	0.10	0.15	0.20	0.30	0.40	0.50	1.00	1.50
0	23.	-1.12	-1.12	-1.29	-1.57					
	24.	-0.74	-0.74	-0.93	-1.18					
	25.	-0.36	-0.36	-0.57	-0.79					
	26.	0.01	0.01	-0.20	-0.40					
	27.	0.38	0.37	0.17	0.00					
	28.	0.75	0.70	0.53	0.39					
	29.	1.11	1.04	0.90	0.79					
	30.	1.46	1.38	1.27	1.19					
0.25	16.	-2.29	-2.29	-2.36	-2.62					
	18.	-1.72	-1.72	-1.83	-2.06	-2.42				
	20.	-1.15	-1.15	-1.29	-1.49	-1.80	2.05	-2.26		
	22	-0.58	-0.58	-0.73	-0.90	-1.17	-1.38	-1.55	-2.17	2.58
	24.	-0.01	-0.01	-0.17	-0.31	-0.53	0.70	-0.84	-1.35	-1.68
	26.	0.56	0.53	0.39	0.29	0.12	0.02	-0.13	-0.52	0.78
	28	1.12	1.06	0.96	0.89	0.77	0.67	0.59	0.31	0.12
	30.	1.66	1.60	1.54	1.49	1.42	1.36	1.31	1.14	1.02
0.50	14.	-1.85	-1.85	-1.94	-2.12	-2.40				
	16.	-1.40	-1.40	-1.50	-1.67	-1.92	2.11	-2.26		
	18.	-0.95	-0.95	1.07	-1.21	-1.43	1.59	-1.73	-2.18	-2.46
	20.	-0.49	0.49	-0.62	-0.75	-0.94	-1.08	-1.20	-1.59	-1.82
	22.	-0.03	-0.03	-0.16	-0.27	-0.43	-0.55	-0.65	-0.98	-1.18
	24.	0.43	0.41	0.30	0.21	0.08	-0.02	-0.10	-0.37	-0.53
	26.	0.89	0.85	0.76	0.70	0.60	0.52	0.46	0.25	0.12
	28.	1.34	1.29	1.23	1.18	1.11	1.06	1.01	0.86	0.77
0.75	14.	-1.16	-1.16	-1.26	-1.38	-1.57	-1.71	-1.82	-2.17	-2.38
	16.	-0.79	-0.79	-0.89	-1.00	-1.17	-1.29	-1.39	-1.70	-1.88
	18.	-0.41	-0.41	-0.52	-0.62	-0.76	-0.87	-0.96	-1.23	-1.39
	20.	-0.04	-0.04	0.15	-0.23	-0.36	-0.45	-0.52	-0.76	-0.90
	22.	0.35	0.33	0.24	0.17	0.07	-0.01	-0.07	-0.27	-0.39
	24	0.74	0.71	0.63	0.58	0.49	0.43	0.38	0.21	0.12
	26.	1.12	1.08	1.03	0.98	0.92	0.87	0.83	0.70	0.62
	28.	1.51	1.46	1.42	1.39	1.34	1.31	1.28	1.19	1.14
1.00	12.	-1.01	-1.01	1.10	-1.19	-1.34	-1.45	-1.53	-1.79	-1.94
	14.	-0.68	-0.68	-0.78	-0.87	-1.00	-1.09	-1.17	-1.40	-1.54
	16	-0.36	-0.36	-0.46	0.53	-0.65	-0.74	-0.80	-1.01	-1.13
	18	-0.04	-0.04	-0.13	-0.20	-0.30	-0.38	-0.44	-0.62	-0.73
	20.	0.28	0.27	0.19	0.13	0.04	-0.02	-0.07	-0.23	-0.32
	22.	0.62	0.59	0.53	0.48	0.41	0.35	0.31	0.17	0.10
	24.	0.96	0.92	0.87	0.83	0.77	0.73	0.69	0.58	0.52
	26.	1.29	1.25	1.21	1.18	1.14	1.10	1.07	0.99	0.94
1.25	10.	-0.90	-0.90	-0.98	-1.06	-1.18	-1.27	1.33	-1.54	-1.66
	12.	-0.62	-0.62	-0.70	-0.77	-0.88	-0.96	-1.02	-1.21	-1.31
	14.	-0.33	-0.33	-0.42	-0.48	-0.58	-0.65	-0.70	-0.87	-0.97
	16.	-0.05	-0.05	-0.13	-0.19	-0.28	-0.34	-0.39	-0.54	-0.62
	18.	0.24	0.22	0.15	0.10	0.03	-0.03	-0.07	-0.20	-0.28
	20.	0.52	0.50	0.44	0.40	0.33	0.29	0.25	0.14	0.07
	22.	0.82	0.79	0.74	0.71	0.65	0.61	0.58	0.49	0.43
	24.	1.12	1.09	1.05	1.02	0.97	0.94	0.92	0.84	0.79
1.50	8.	-0.82	-0.82	-0.89	-0.96	-1.06	-1.13	-1.19	-1.36	-1.45
	10.	-0.57	-0.57	-0.65	-0.71	-0.80	-0.86	-0.92	1.07	-1.16
	12.	-0.32	-0.32	-0.39	-0.45	-0.53	-0.59	-0.64	-0.78	-0.85
	14.	-0.06	-0.07	-0.14	-0.19	-0.26	-0.31	-0.36	-0.48	-0.55
	16.	0.19	0.18	0.12	0.07	0.01	-0.04	-0.07	-0.19	0.25
	18.	0.45	0.43	0.38	0.34	0.28	0.24	0.21	0.11	0.05
	20.	0.71	0.68	0.64	0.60	0.55	0.52	0.49	0.41	0.36
	22.	0.97	0.95	0.91	0.88	0.84	0.81	0.79	0.72	0.68

Table 3.3: (Continued)

Activity Level 100 kcal/m^2hr

Clothing clo	Ambient Temp. °C	Relative Velocity (m/s) <0.10	0.10	0.15	0.20	0.30	0.40	0.50	1.00	1.50
0	18.		−2.00	−2.02	−2.35					
	20.		−1.35	−1.43	−1.72					
	22.		−0.69	−0.82	−1.06					
	24.		−0.04	−0.21	−0.41					
	26.		0.59	0.41	0.26					
	28.		1.16	1.03	0.93					
	30.		1.73	1.66	1.60					
	32.		2.33	2.32	2.31					
0.25	16.		−1.41	−1.48	−1.69	−2.02	2.29	−2.51		
	18.		−0.93	−1.03	−1.21	−1.50	−1.74	1.93	2.61	
	20.		−0.45	−0.57	−0.73	−0.98	−1.18	−1.35	−1.93	−2.32
	22.		0.04	−0.09	−0.23	−0.44	−0.61	−0.75	−1.24	−1.56
	24.		0.52	0.38	0.28	0.10	−0.03	−0.14	−0.54	−0.80
	26.		0.97	0.86	0.78	0.65	0.55	0.46	0.16	−0.04
	28.		1.42	1.35	1.29	1.20	1.13	1.07	0.86	0.72
	30.		1.88	1.84	1.81	1.76	1.72	1.68	1.57	1.49
0.50	14.		−1.08	−1.16	−1.31	−1.53	−1.71	−1.85	−2.32	
	16.		−0.69	−0.79	−0.92	−1.12	1.27	−1.40	−1.82	−2.07
	18		−0.31	−0.41	−0.53	−0.70	0.84	−0.95	−1.31	−1.54
	20.		0.07	−0.04	−0.14	−0.29	−0.40	−0.50	−0.81	−1.00
	22.		0.46	0.35	0.27	0.15	0.05	−0.03	−0.29	−0.45
	24		0.83	0.75	0.68	0.58	0.50	0.44	0.23	0.10
	26.		1.21	1.15	1.10	1.02	0.96	0.91	0.75	0.65
	28.		1.59	1.55	1.51	1.46	1.42	1.38	1.27	1.21
0.75	10.		−1.16	−1.23	−1.35	−1.54	−1.67	−1.78	−2.14	−2.34
	12.		−0.84	−0.92	−1.03	−1.20	−1.32	−1.42	−1.74	−1.93
	14.		−0.52	−0.60	−0.70	−0.85	−0.97	−1.06	−1.34	−1.51
	16.		−0.20	−0.29	−0.38	−0.51	−0.61	−0.69	−0.95	−1.10
	18.		0.12	0.03	−0.05	−0.17	−0.26	−0.32	−0.55	−0.68
	20.		0.43	0.34	0.28	0.18	0.10	0.04	−0.15	−0.26
	22.		0.75	0.68	0.62	0.54	0.48	0.43	0.27	0.17
	24.		1.07	1.01	0.97	0.90	0.85	0.81	0.68	0.61
1.00	8.		−0.95	−1.02	−1.11	−1.26	−1.36	−1.45	−1.71	−1.86
	10.		−0.68	−0.75	−0.84	−0.97	−1.07	−1.15	−1.38	−1.52
	12.		−0.41	−0.48	−0.56	−0.68	−0.77	−0.84	−1.05	−1.18
	14.		−0.13	−0.21	−0.28	−0.39	−0.47	−0.53	−0.72	−0.83
	16.		0.14	0.06	0.00	−0.10	−0.16	−0.22	−0.39	−0.49
	18.		0.41	0.34	0.28	0.20	0.14	0.09	−0.06	−0.14
	20.		0.68	0.61	0.57	0.50	0.44	0.40	0.28	0.20
	22.		0.96	0.91	0.87	0.81	0.76	0.73	0.62	0.56
1.25	−2.		−1.74	−1.77	−1.88	−2.04	−2.15	−2.24	−2.51	−2.66
	2.		−1.27	−1.32	−1.42	−1.55	−1.65	−1.73	−1.97	−2.10
	6.		−0.80	−0.86	−0.94	−1.06	−1.14	−1.21	−1.41	−1.53
	10.		−0.33	−0.40	−0.47	−0.56	−0.64	−0.69	−0.86	−0.96
	14.		0.15	0.08	0.03	−0.05	−0.11	−0.15	−0.29	−0.37
	18.		0.63	0.57	0.53	0.47	0.42	0.39	0.28	0.22
	22.		1.11	1.08	1.05	1.00	0.97	0.95	0.87	0.83
	26.		1.62	1.60	1.58	1.55	1.53	1.52	1.47	1.45
1.50	−4.		−1.52	−1.56	−1.65	−1.78	−1.87	−1.95	−2.16	−2.28
	0.		−1.11	−1.16	−1.24	−1.35	−1.44	−1.50	−1.69	−1.79
	4.		−0.69	−0.75	−0.82	−0.92	−0.99	−1.04	−1.20	−1.29
	8.		−0.27	−0.33	−0.39	−0.47	−0.53	−0.58	−0.72	−0.79
	12.		0.15	0.09	0.05	−0.02	−0.07	−0.11	−0.22	−0.29
	16.		0.58	0.53	0.49	0.44	0.40	0.37	0.28	0.23
	20.		1.01	0.97	0.94	0.91	0.88	0.85	0.79	0.75
	24.		1.47	1.44	1.43	1.40	1.38	1.36	1.32	1.29

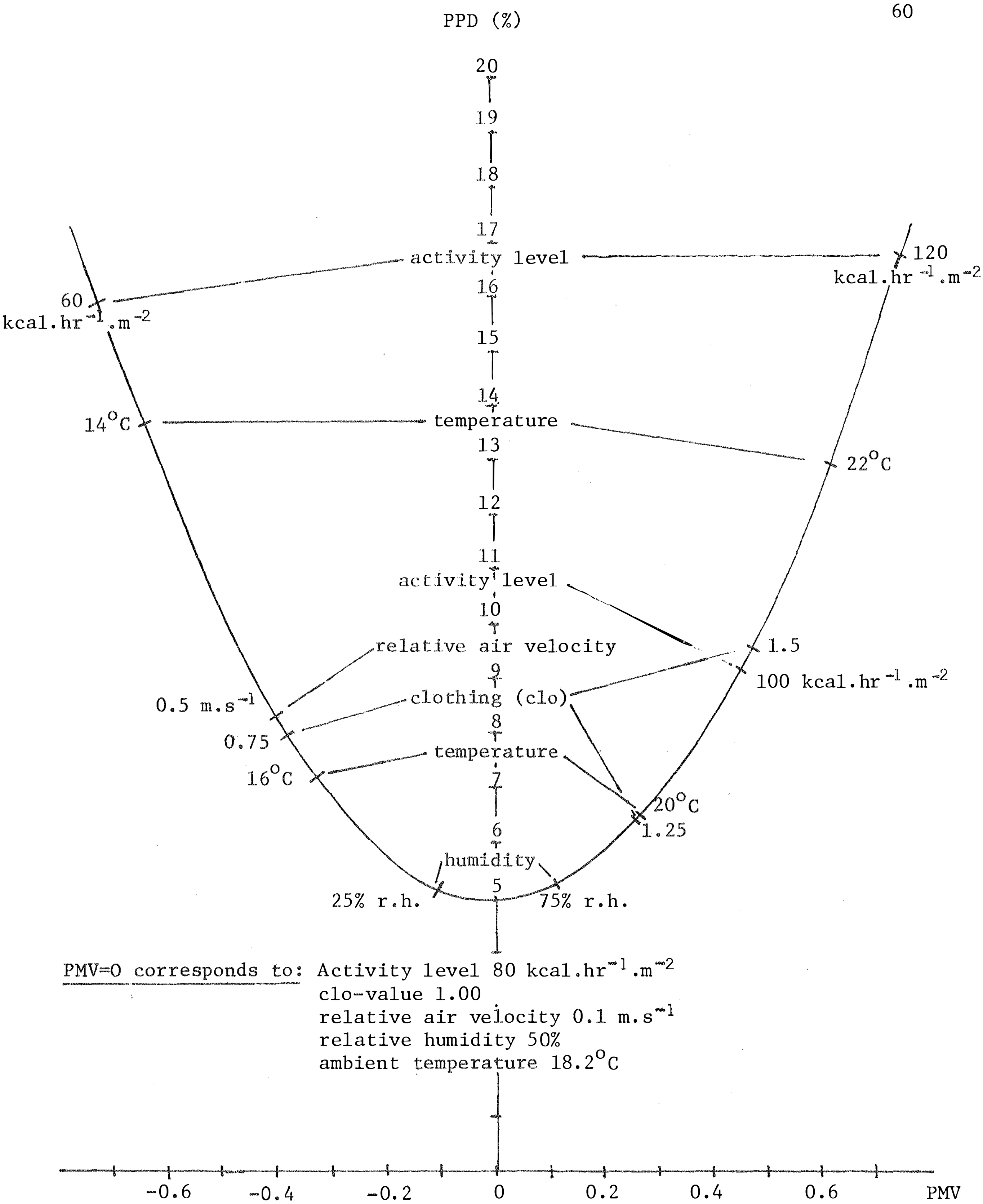


Figure 3.3: Effects on comfort (PMV) and discomfort (PPD) of independent changes in the thermal variables

Temperature

Perceived temperature actually results from the combined effects of three components, corresponding to heat transfer by conduction, convection and radiation. The relevant parameters are, respectively, floor temperature, air temperature (t_a) and mean radiant temperature (t_{mrt}).

Floor Temperature

Muncey (1954) and Muncey and Hutson (1953) have shown that the temperature and materials of flooring have little effect on the thermal comfort of people wearing normal footwear. This is not really surprising, considering the insulating value of footwear. They argue that the feeling of "cold feet" results from the body's normal mechanism for conserving heat by reducing the flow of blood to its extremities. Cold feet can be avoided by increasing the overall temperature of the body's surroundings, rather than just the temperature around the feet. For a person in bare feet, foot comfort depends on the rate of heat loss from the feet to the floor. This rate, in turn, depends on the temperature and thermal properties of the floor. Fanger (1970) has estimated comfort limits for some flooring materials. These are shown in Table 3.4.

Table 3.4: Estimated comfort temperature ranges for different flooring materials, people in bare feet (Fanger, 1970).

Flooring material (without surface finishing)	Comfort range of floor temperature (^{o}C)
steel	29 - 32
concrete	27 - 34
linoleum	24 - 35
oak wood	22 - 35
pine wood	17 - 39
cork	5 - 42

Radiant Temperature

The effects of radiant heating from sunshine and heaters are widely appreciated. Objects at room temperatures also emit radiant heat. The effects of this heat are rather less dramatic, for two reasons. Firstly, heat is radiated at a rate proportional to the fourth power of the absolute temperature. Thus, a radiator bar at room temperature ($20^{o}C$, or about 300 K) emits only 0.4 per cent of the radiant heat emitted when it is turned on, at around $900^{o}C$ (1200 K). Secondly, the human body itself radiates heat. When the body is at the same temperature as its surroundings, the *net* radiant heat exchange is zero.

Mean radiant temperature (t_{mrt}) refers to the uniform temperature of black surroundings which would produce the same net radiant heat exchange as the surroundings in question. Temperatures of the walls, floor and roof of a room usually vary. When the variation is small (only a few degrees), t_{mrt} can be estimated by weighting surface temperatures according to area, and finding the mean of the weighted temperatures.

When a heater with an appreciable radiant heat output is used, the mean radiant temperature can be calculated by the formula:

$$T_{mrt} = (T^{4}_{umrt} + 0.208 \cdot 10^{8}\ f_p\ \alpha_{ir}\ q_{ir})^{0.25}$$

where: T_{umrt} = unirradiated mean radiant temperature (K), i.e. the mean radiant temperature of the surroundings, excluding the heater.

f_p = projected area factor of the person. This factor varies with posture (it is usually lower for a seated than for a standing person) and with the angle of the heater relative to the person. 0.25 is a typical value. A complete set of values is given by Fanger (1970).

α_{ir} is the absorptance of the outer surface of the person for the incident radiation. For a heat source at 1200 K (atmospheric gas-fired radiant heater), α_{ir} = 0.9 for a person in medium-grey clothing. The absorptance does not vary greatly with the colour of clothing, although it decreases for higher temperature radiant heat sources (α_{ir} = 0.8 @ 2500 K).

q_{ir} = the radiant flux density produced by the heater at the position of the person ($kcal.m^{-2}.hr^{-1}$). This depends on the heater's radiant output and the angle of the person relative to the heater. It also decreases according to the square of the distance between the person and the heater.

Air Temperature

The rate of heat conduction through the clothing is proportional to the difference between the air temperature (t_a) and the skin temperature. There is also a respiration heat loss due to air inhaled (at t_a) being heated to approximately 34°C before being exhaled.

Ambient Temperature

Both mean radiant temperature and air temperature have important effects on thermal comfort. The ambient temperatures used in Table 3.3 refer specifically to conditions under which air temperature and mean radiant temperature are equal. When t_a is not equal to t_{mrt} an approximate ambient temperature is given by the mean of t_{mrt} and t_a (ASHRAE *Handbook of Fundamentals*, 1972). This is valid for a nude, sedentary body in still air. At higher activity levels, wind speeds and clothing resistances, t_{mrt} becomes less important. To maintain comfort at high activity (150 $kcal.m^{-2}.hr^{-1}$), in heavy clothing (clo = 1.5) at a wind velocity of 5 $m.s^{-1}$, a change of 1°C in t_a requires a compensating change of 6°C in t_{mrt}. Under normal conditions (Figure 3.3) a 1°C change in t_a corresponds to a 1.2°C change in t_{mrt}.

From Figure 3.3, it can be seen that the difference between "slightly cool" (PMV = -1) and "slightly warm" (PMV = +1) corresponds to a range of ambient temperatures from 12°C to 24°C.

Fanger (1970) concludes that people can tolerate quite large asymmetries in radiant fields without discomfort, provided they are in overall comfort. It has been stated (Constance, 1978) that a feeling of stuffiness occurs when the head is more than 3°C warmer than the feet. No reference is given, and it is not clear whether the temperature referred to is ambient temperature or skin temperature. Nevertheless, this agrees qualitatively with everyday experience, and implies that vertical temperature gradients cause stuffiness.

3.3 HEATING STANDARDS FOR TASMANIAN HOMES

> "Every schoolroom should have a thermometer hung *at the breathing line*, and well away from the fire. The temperature should be noted during each lesson, and the room regulated as nearly as possible to 60°F. It should never be more than 66°F or less than 55°F in school hours."
>
> – Extract from "The Warming of Schoolrooms" by Dr. Elkington, (*Educational Record* (Tas.), May, 1906).

Using the information described in section 3.2, a modern heating standard (Table 3.5) can be derived according to specified thermal parameters. As with the standards used by Coldicutt *et al.* (1978) and Shaw and Stephenson (1977) a multiple standard is used, relating to the living and sleeping areas of the house. The thermal parameters for the two daytime standards are listed in Table 3.5, and the relationship between t_{mrt} and t_a, for each standard, is shown in Figure 3.4.

Both standards apply to daytime use. At night, overall thermal comfort can be maintained, without reference to ambient temperature, by using sufficient blankets, or supplementing metabolic heat (which falls to 35 $kcal.m^{-2}.hr^{-1}$ during sleep) with heat from an electric blanket. Discomfort may occur at low ambient temperatures due to excessive heat loss from the exposed face. Fanger does not provide enough information to quantify this condition. Shaw and Stephenson (1977) have assumed a standard of 15°C for sleeping areas. Coldicutt *et al.* (1978) take the acceptable range of environmental temperatures in the sleeping area to be 13°C to 21°C. Both of the derived comfort standards fall within this range. The lower standard (B) will be applied to night-time use.

Coldicutt *et al.* (1978) have shown that a small reduction in heating temperature (from 21°C to 18°C) can result in a considerable annual saving of energy (from 33 GJ down to 20.1 GJ). Energy conservation measures taken in the U.S.A. have made use of this fact by turning down the thermostats in office buildings. A change from optimum, however, increases thermal discomfort, as measured by PPD. In a home, the situation is somewhat simpler: the thermostat can be turned down until

Table 3.5: Thermal comfort conditions for Tasmanian homes.

	STANDARD A (Living Zone)		STANDARD B (Sleeping Zone)	
Thermal Parameter	Value	Description	Value	Description
Activity level ($kcal.m^{-2}.hr^{-1}$)	50	seated, quiet	100	average housework
Clothing (clo)	1.5	winter clothing, seated in upholstered chair.[1]	1.15	winter clothing
Relative air velocity ($m.s^{-1}$)	< 0.1	still air	0.1	air velocity due to minor body movements
Relative humidity[2]	50%		50%	
$\frac{\delta t_a}{\delta t_{mrt}}$ [3]	- 0.9		- 0.8	
Ambient temperature for comfort	20.2°C		13.6°C	

Notes:

1. *Clothing:* the clo-value of 1.5 includes an allowance for the large thermal resistance provided to approximately one-third of the body by an upholstered armchair or couch.

2. *Relative humidity:* Hobart's relative humidity averages 70-75 per cent in winter (*Tasmanian Year Book*, 1977). This would result in temperature corrections of -0.7°C for Standard A and -0.5°C for Standard B. In Launceston, the humidity is 5 ∿ 10 per cent higher, resulting in corrections of -0.8°C and -0.6°C.
   It is assumed, however, that the higher temperatures in a heated area result in a lowering of relative humidity to approximately 50 per cent. In that case, no corrections would be required.

3. $\frac{\delta t_a}{\delta t_{mrt}}$: this expresses the relative importance of air temperature and mean radiant temperature for each standard (Figure 3.4).

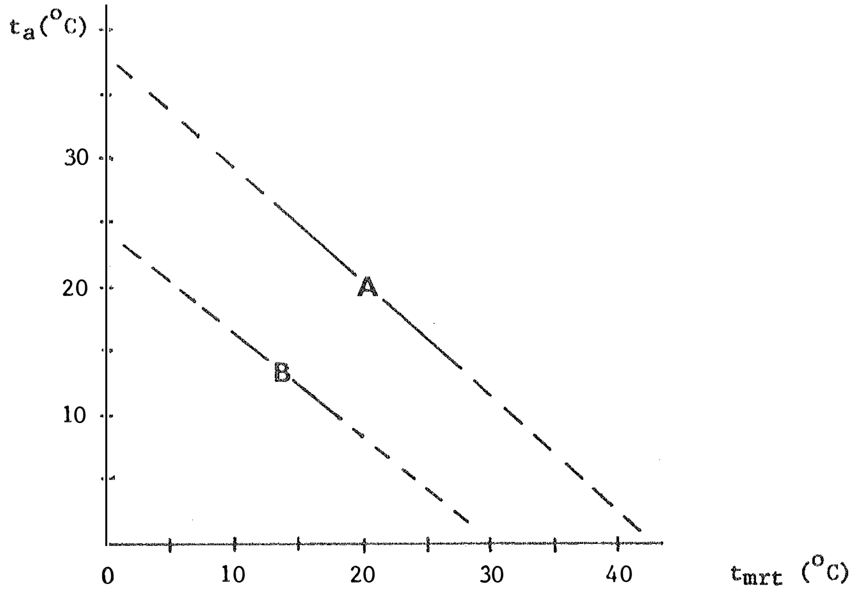


Figure 3.4: Air temperatures (t_a) and mean radiant temperatures (t_{mrt}) for comfort standards A and B.

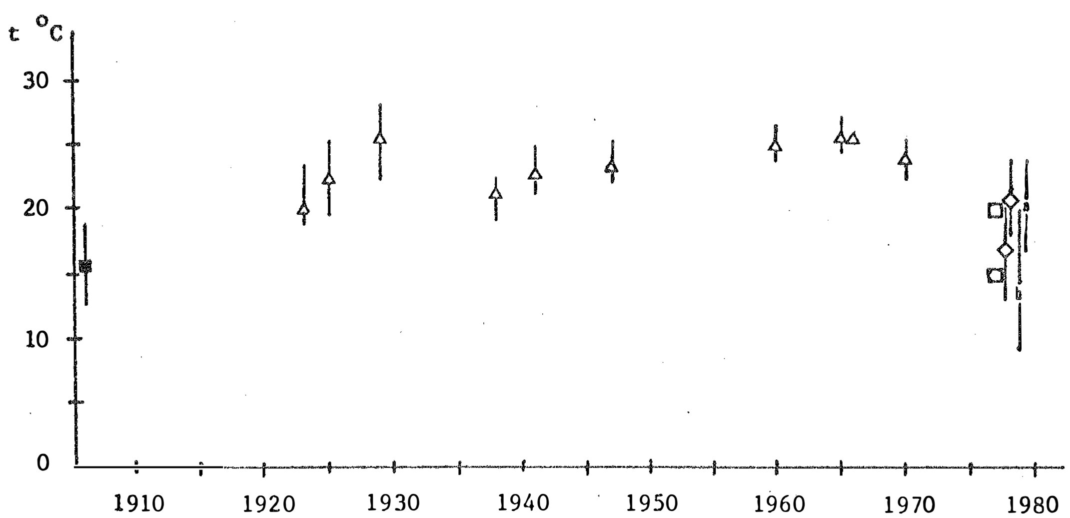


Figure 3.5: Comfort temperature standards, 1906-1979

Sources:	Elkington (1906):	■	(Tas.)
	McNall (1973):	△	(U.S.A.)
	Shaw and Stephenson (1977):	□	(N.Z.)
	Coldicutt *et al.* (1978):	◇	(Tas. and Vic.)
	Comfort Standard A:	a	(Tas.)
	Comfort Standard B:	b	(Tas.)

one member of the family experiences discomfort. For a group of five people, this point can be estimated by a PPD of 20 per cent.

Using this criterion, a family can be expected to experience thermal comfort over a range of temperatures from 17°C to 24°C (Standard A) and from 9°C to 20°C (Standard B).

It is instructive to compare these standards with other temperature standards that have been derived over time. Figure 3.5 depicts a number of temperature standards dating from 1906 to the present.

The following points emerge:

(i) The United States' temperatures are higher than those for Australia and New Zealand. This is due to the fact that they apply primarily to office work (i.e. clo-value ∿ 1.0, relative humidity 40 per cent, activity level 50-60 $kcal.m^{-2}hr^{-1}$) rather than domestic situations.

(ii) Allowing for point (i), temperatures have tended to increase over time. McNall (1973) attributes this partly to a change in fashion, away from heavy clothing (such as long woollen underwear), and partly to reduced activity levels in the workplace. Since summer cooling standards have remained constant at 25.6°C, further increases in heating levels are unlikely.

The comfort standards need only apply while the house or room is occupied. Usually, a large amount of heating can be saved by not heating during unoccupied hours. This applies particularly to areas heated to higher temperatures. Suggested heating requirements are shown, for all the areas of a house, in Figures 3.6 and 3.7. Figure 3.6 applies to a house in which the whole family is absent during working and school hours. Figure 3.7 applies to a house that is occupied throughout the day.

Because of the irregular nature of its use, the bathroom is usually heated independently, either by radiant or forced convection heaters.

The major heating appliance in a Tasmanian home is located in the living room, and usually serves to heat the dining area and kitchen. Few of these appliances are thermostatically controlled. Coldicutt *et*

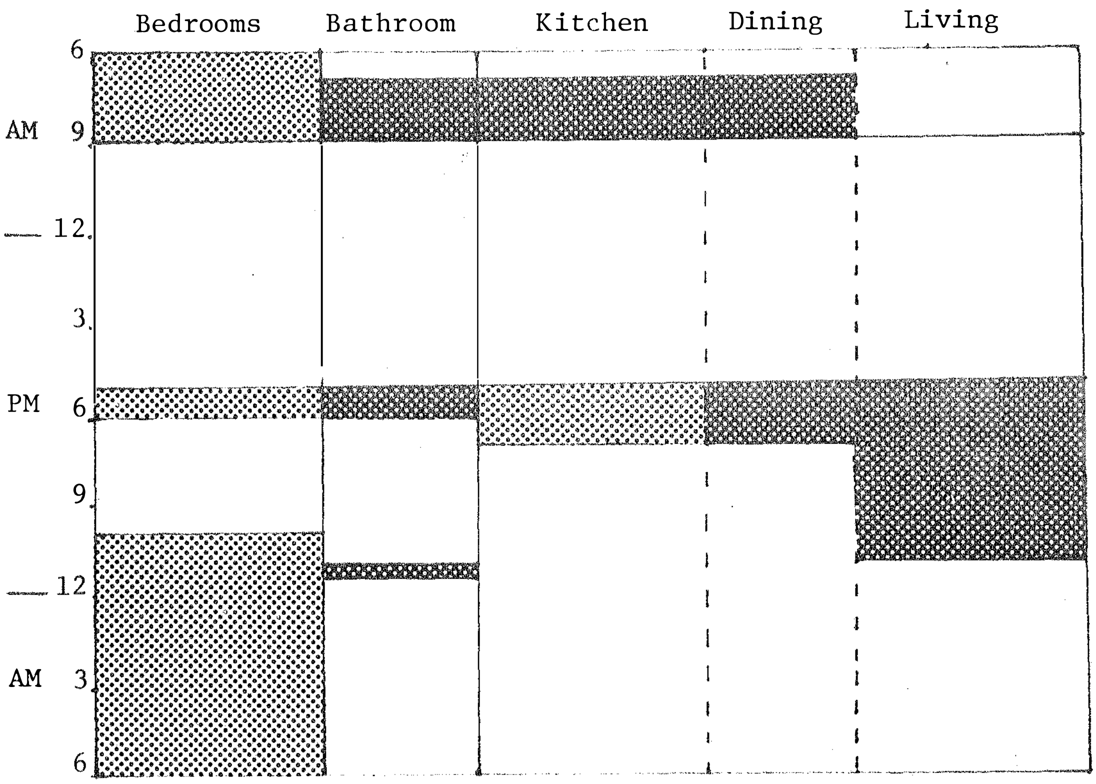


Figure 3.6: Heating requirements of a house not occupied during working hours

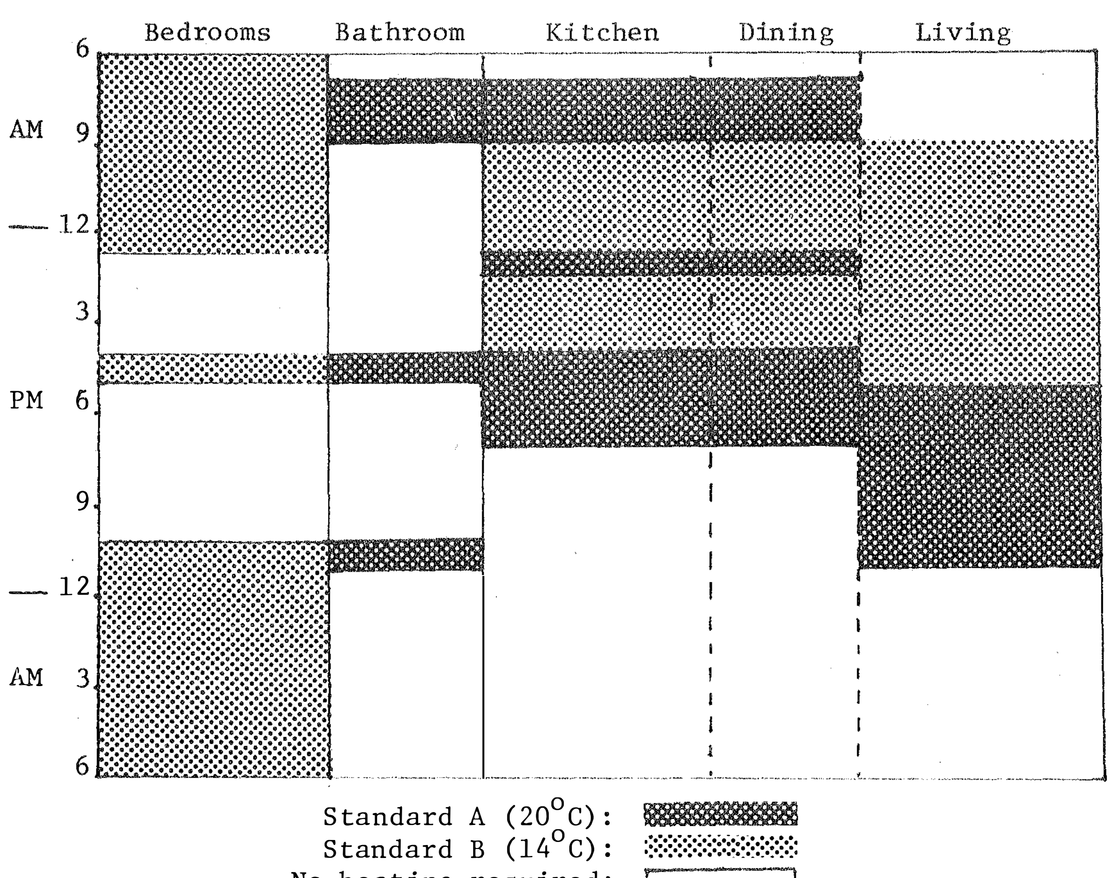


Figure 3.7: Heating requirements of a house occupied throughout the day

energy use out of proportion to the improvement in comfort.

There appear to be three feasible solutions to the problem of warm-up:

(i) the supplementary resistance heating could be used to boost the heat pump output during warm-up;

(ii) radiant heating could be used in conjunction with the heat pump. Radiant heaters would provide an instant increase in radiant temperature, and could be used in place of the normal supplementary heating. In the latter case, the higher radiant temperature would mean a lower air temperature for comfort. A compensating switch could adjust the thermostat automatically when the radiant heaters were operating;

(iii) a time switch could be used to pre-heat the house for half to one hour prior to the expected return.

Time switches have recently begun appearing on the more sophisticated electric convection heaters. Their benefits in terms of comfort and/or cost savings seem to outweigh the extra cost involved. Ideally, a time-clock could be pre-set to provide the desired heating patterns to each zone, on a seven-day cycle, and the unit could be packaged with the thermostat and a clock, for decorative wall-mounting.

It is interesting to compare this proposed heating standard with average heating use in Tasmania. Coldicutt *et al.* (1978) have estimated that an uninsulated brick veneer house in Hobart can have its living zone (lounge, kitchen and passage) heated to 18^{o}C during the day and evening at an energy cost of 35 GJ per year.

Hartley, Jones and Badcock (1978) have estimated Tasmania's average heat use per dwelling to be 28.43 GJ in 1975. However, the 40.4 per cent of houses with oil heaters (A.B.S., 1978) consume 55.9 per cent of the heating energy, or 39.3 GJ per household. Thus, it appears that people with oil heating approach the proposed standard, at least for their living zone. The remainder of the population, in general, appears to be tolerating a significantly lowered standard of thermal comfort.

The amount of heating energy used per dwelling increased steadily from

13.36 GJ per annum in 1967 to 28.43 GJ per annum in 1975, indicating that overall heating standards are rising rapidly.

3.4 TASMANIA'S CLIMATE AND HEATING REQUIREMENTS

The subject of cooling for comfort has been largely ignored until now. In fact, because Hobart's outdoor temperature exceeds $25^{\circ}C$ for only 96 hours per year, there is little justification for installing an air-conditioner for cooling purposes.

The heating requirement, on the other hand, is among the highest in Australia (though not as great as found in some other parts of the world). Table 3.6 shows the heating requirements, in terms of *degree-days*, of a number of cities, including Hobart.

The degree-day figure is calculated according to the formula:

$$\text{degree days (d.d.)} = \sum_{365\ \text{days}} \Delta t$$

$$\begin{cases} \Delta t = 0, \text{ if } \tfrac{1}{2}(t_{max} + t_{min}) > t_b \\ \Delta t = t_b - \tfrac{1}{2}(t_{max} + t_{min}), \text{ if } \tfrac{1}{2}(t_{max} + t_{min}) < t_b \end{cases}$$

(t_{max} and t_{min} are the daily maximum and minimum temperatures, respectively. T_b is the selected base temperature.)

For calculations of heating requirements, the choice of a base temperature is important. The base temperature is the outdoor air temperature at which the heat losses from a house at comfort temperature are just balanced by the average heat gains from solar radiation, appliances and occupancy (see Chapter Four for details).

The annual heat input required to maintain a house at the comfort temperature can be calculated from the degree-day figure:

$$\text{Annual heat requirement (kWh)} = \text{d.d.} \times \frac{24}{1000} \times U_{total}$$

where U_{total} is the total rate of heat loss (conduction + convection) from the house, in terms of watts per degree centigrade.

The degree-day method has been widely used for estimating heating requirements. However, it has a number of drawbacks, some of which are

Table 3.6: Average annual heating degree days (base 18.3^{o}C) for a number of cities. [Rawlings *et al.* (1977); O'Brien, (1976); Coldicutt *et al.*(1978)].

Locality	Heating degree-days (base 18.3^{o}C)
Moscow	5360
Copenhagen	3835
London	2947
New York	2718
Ballarat	2386
Hobart	2305
Canberra	2275
Colac	2251
Yallourn	1959
Bendigo	1817
Essendon	1796
Geelong	1667
Melbourne	1490
Adelaide	1280
Perth	775
Sydney	732
Brisbane	312
Townsville	43
Darwin	0

serious:

(i) The mean temperature may differ from the average of maximum and minimum temperatures. Daily temperature profiles for Hobart (Figure 3.8) suggest that the temperature remains near its minimum value for slightly longer than near its maximum. Analysis of temperatures for 59 days (the fifteenth day of each month) over five years (1973-1977) show that the hourly mean temperature averages $0.17^{\circ}C$ lower than the average of the maximum and minimum. This means that the degree-day figure for Hobart is overestimated by about 45 degree-days (i.e. 2 per cent).

(ii) The base temperature will depend on the thermal properties of the house in question. For conventional, uninsulated houses this variation will be small. Well-insulated houses have the same heat gains, from appliances and occupants, as uninsulated houses, and their solar heat gains are almost as great. These heat gains will have a greater effect on the temperature inside an insulated house. So insulation affects not only the specific rate of heat loss, but also the base temperature. For each base temperature, a new degree-day figure must be calculated from the temperature figures.

(iii) The degree-day figure takes no account of extreme temperatures. Hobart's cooling requirement (calculated in a similar way to the heating degree-days) is 2½ degree-days. Calculated on an hourly basis, it is 328 degree-hours (13.7 degree-days). The discrepancy is due to the fact that high temperatures occur for short periods on many days whose average temperatures are below $25^{\circ}C$. Short periods of extreme temperatures will have little effect on heavily constructed or well insulated buildings, but most uninsulated houses respond to outdoor temperatures in 1 to 3 hours. Hobart has only a small daily temperature swing of $5-6^{\circ}C$. The effects of this swing on heating needs will be less than those found in other places.

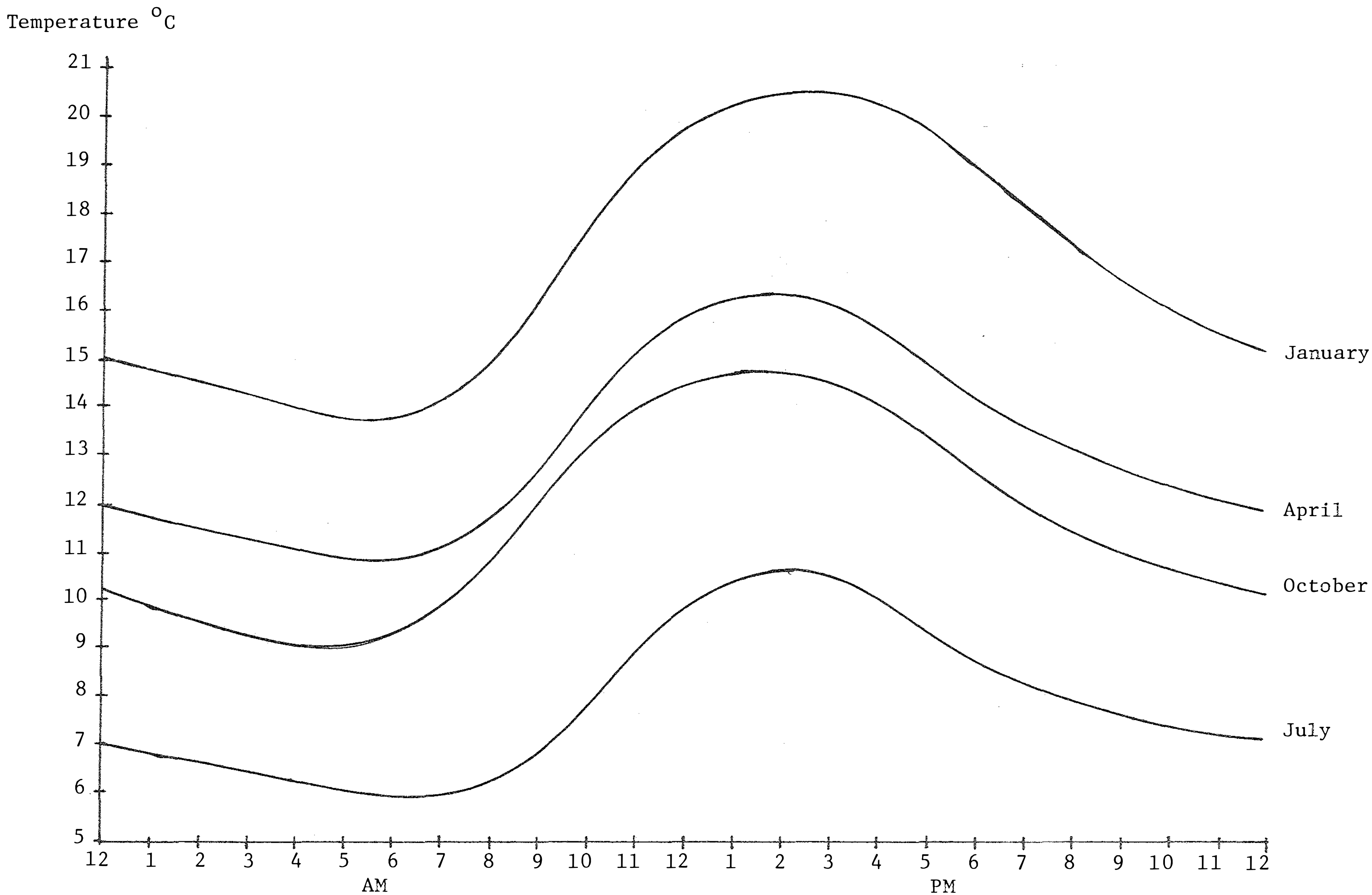


Figure 3.8: Mean hourly temperatures, °C (Hobart, 1973-77) for January, April, July and October

(iv) Degree-day calculations are only valid for continuous heating. If heating is needed for only part of the day, as described in section 3.3, then the amount of heating required depends very much on *when* it is needed. Obviously (Figure 3.8), daytime heating will require less energy than heating at night.

(v) The degree-day figure cannot be used to predict the power requirements of an air-source heat pump. The COP of an air-source heat pump relates directly to the air temperature, and is lower at low temperatures when the heat requirements are greatest. Consequently, the overall COP will be lower than the COP suggested by the average temperature for the day.

For these reasons, it is preferable to use a calculation method based on hourly temperatures, when they are available. The volume of calculation required demands the use of a computer. The TEMPAL computer package was found to have the ability to perform most of the calculations required. A table of the heating requirements over the range of combinations of outdoor temperature and heating load was added to enable calculation of COP's.

Predicting the Climate

From a limited amount of data (7 years) available on magnetic tape for computer use, Coldicutt *et al.* (1978) have chosen the winter of 1970 as the representative heating season for Hobart. The heating season is taken over nine months, from March to November. Air temperature, solar radiation, wind speed, cloud cover and relative humidity data from the Hobart recording station are used to calculate indoor temperatures of the houses studied.

The use of such information for predicting the performance of a heat pump over the decade from 1980-1990 raises a number of quite valid questions. The first question is: "Can we assume that conditions in the future will be the same as conditions in the past?"

Discussion of this question will be confined here to temperature, which is the dominant climatic factor affecting heating requirements.

Analysis of mean annual and mid-winter (July) maximum and minimum temperatures (Figure 3.9) shows a statistically significant increase in annual average temperatures. The minimum temperature is increasing at 0.034^{o}C per year, and the maximum temperature at 0.008^{o}C per year. These trends are both statistically significant at the 5 per cent level. Increases in temperatures for July are not statistically significant. If the trend in mean annual minimum temperatures affects those throughout the heating season, the annual heating requirement will fall by about 80 degree-days (3.5 per cent) over a decade.

The second question is: "How well does the chosen period represent conditions over a longer period?" A complete answer to this question would require more calculations than are saved by choosing a representative heating season. The 1970 heating season was chosen after examining monthly average and extreme conditions over the seven-year period for which data were available. It was chosen so that the total energy consumption would come out as an average value.

Some idea of the heating requirements can be obtained using degree-day estimates calculated from the mean hourly temperatures of each month. In Figure 3.10, monthly degree-day estimates for the 1970 heating season are compared with figures averaged over ten years (1968-1977). The most notable departures from average occur in August, September and October, which all had above average degree-day estimates. However, the most severe month (August) had a degree-day estimate only slightly higher than the most severe average month (July). The winter is better described as prolonged, rather than extreme. The degree-day estimate for the nine months was 2140, compared with 2228 for the twelve average months (and 2305 by the conventional method).

Thus, heating requirements given for the 1970 heating season probably underestimate average requirements by about 4 per cent.

Application of Results to Other Localities

Strictly speaking, the climatic data apply only to the meteorological recording station in Hobart. Figure 3.11 demonstrates that temperatures *within the locality of the station* may vary by several degrees. The variation shown in the Figure is due primarily to the formation of what is known as a heat island, in the central business district of Hobart.

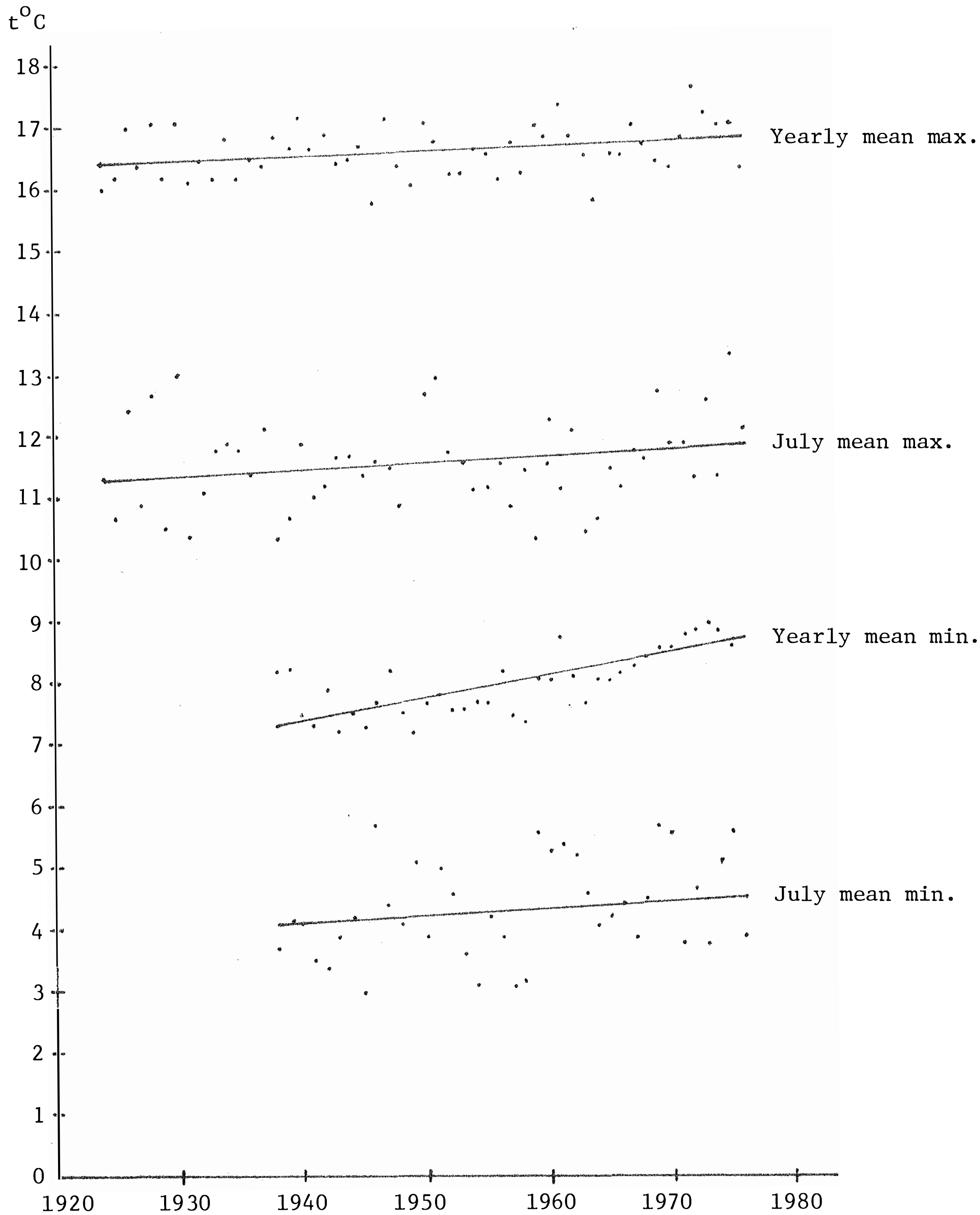


Figure 3.9: Trends in maximum and minimum temperatures for Hobart: Mean annual and mean July temperatures.

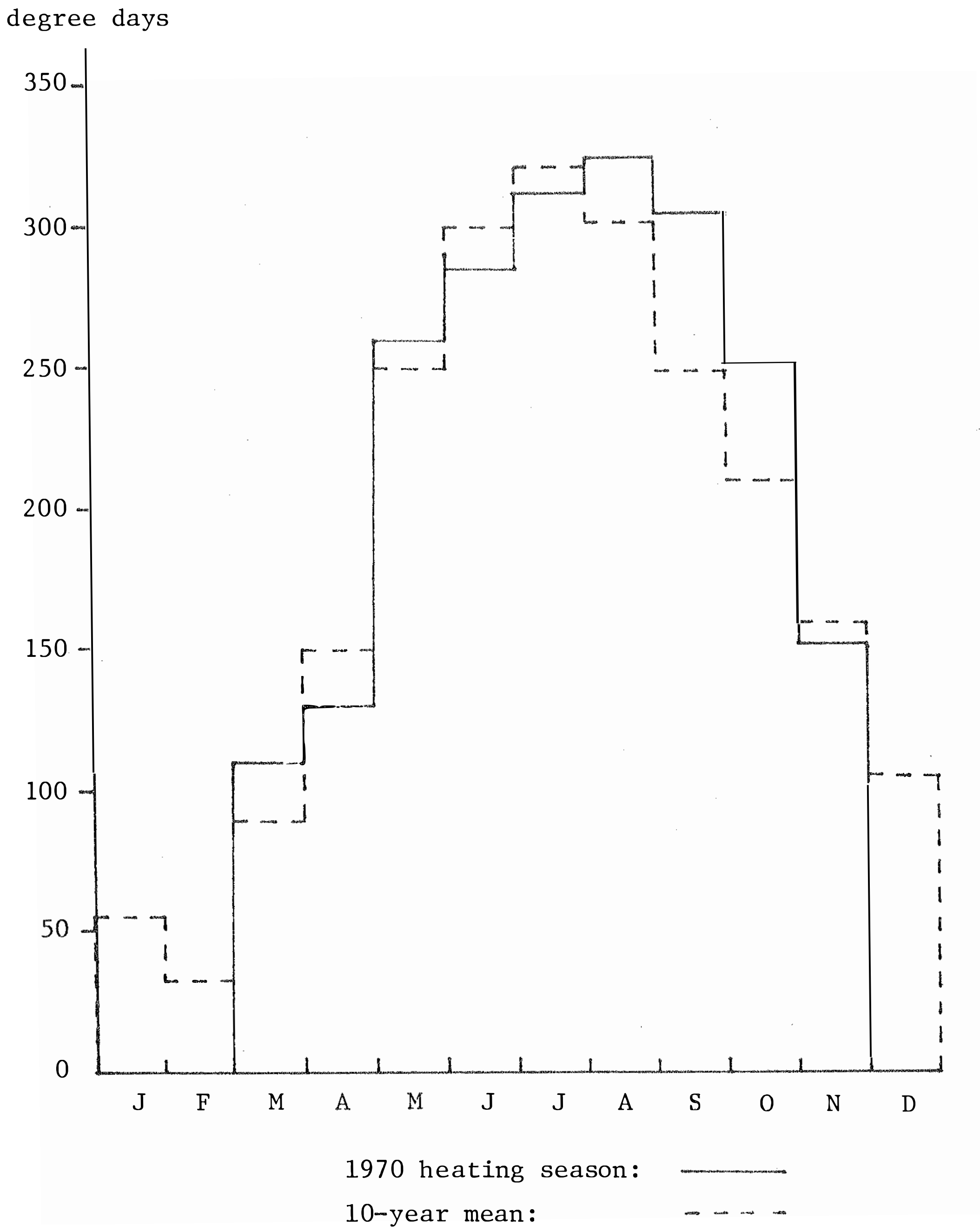


Figure 3.10: Comparison of 1970 heating season with 10-year mean (degree-days estimated from monthly mean temperatures to base 18.3^{o}C)

The elongated area of low temperature (centre left) is due to cold air drainage down a valley. The shape of the heat island pattern varies in time, and according to weather conditions. In cloudy conditions, the variation is expected to be lower. The recording station is on the fringes of the heat island, and so could be expected to record temperatures as much as 1^{o}C higher (overall) than those experienced in the residential areas of Hobart. Annual heating requirements in the colder areas of Hobart could be underestimated by more than 10 per cent.

Ideally, heating requirements for localities other than Hobart should be calculated using local climatic data. However, hourly climatic data are only collected continuously at three locations (Hobart, Hobart airport and Launceston airport) in Tasmania. The only urban location of the three is Hobart. Observations are made at three-hour intervals at some other locations, but these are not adequate for accurate computer estimations. Daily maximum/minimum temperatures are recorded in most areas.

Due to the lack of more detailed information, estimating heating requirements for places other than Hobart reduces to a choice between degree-day calculations, and extrapolating from the Hobart results. Apart from its other limitations, the degree-day method would require calculations of degree-days for each base temperature at each location. To avoid this problem, results will be extrapolated from Hobart to other places.

The economic viability of heat pumps can be determined from the ratio:

$$\frac{\text{net value of savings}}{\text{net value of costs}}$$

The savings will be approximately proportional to the annual heating demands. The savings will depend slightly on the temperatures at which heating occurs, since the COP depends on temperature.

The costs are related primarily to the required capacity of the unit. Normally, a *design temperature* is used, to choose the heating capacity according to the difference between it and the base temperature. The outdoor temperature only falls below the design temperature for a small percentage (usually 2.5 per cent or 5 per cent) of the time, but these short periods of discomfort are compensated for by a reduction in the

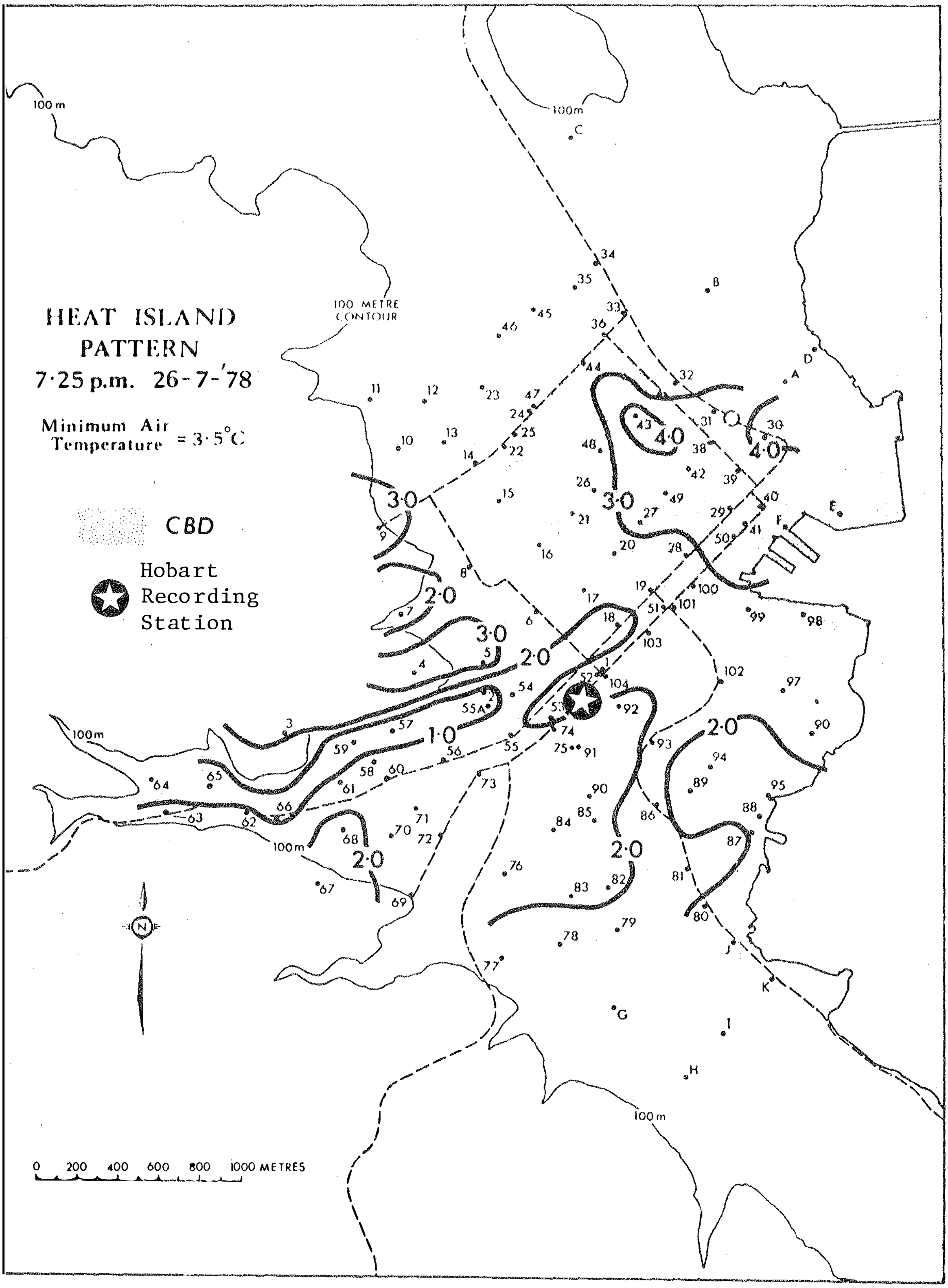


Figure 3.11: Temperature distribution in Hobart on a clear winter's night. (Source: M. Nunez, Geography Department, University of Tasmania)

initial cost of the heater. Design temperatures have been estimated for Hobart (2.8^{o}C, Woolridge, 1976), Hobart airport (3^{o}C, Drysdale, 1959) and Launceston airport (3^{o}C, Drysdale, 1959).

The best readily available estimate for a design temperature is the mean minimum temperature for the coldest month. Use of the degree-day figure for comparison of climates (as opposed to estimating heating loads) avoids many of its limitations. The ratio of savings to costs can be estimated by multiplying the Hobart ratio by the climatic adjustment factor:

$$\text{climatic adjustment factor} = 6.4 \times 10^{-4} \frac{d.d.}{(18.3 - t_{min})}$$

where: 6.4×10^{-4} is a normalising constant chosen so that the climatic adjustment factor for Hobart is unity.

d.d. is the degree-day estimate (base 18.3^{o}C) for the location being considered (estimated from monthly mean maximum/minimum temperatures).

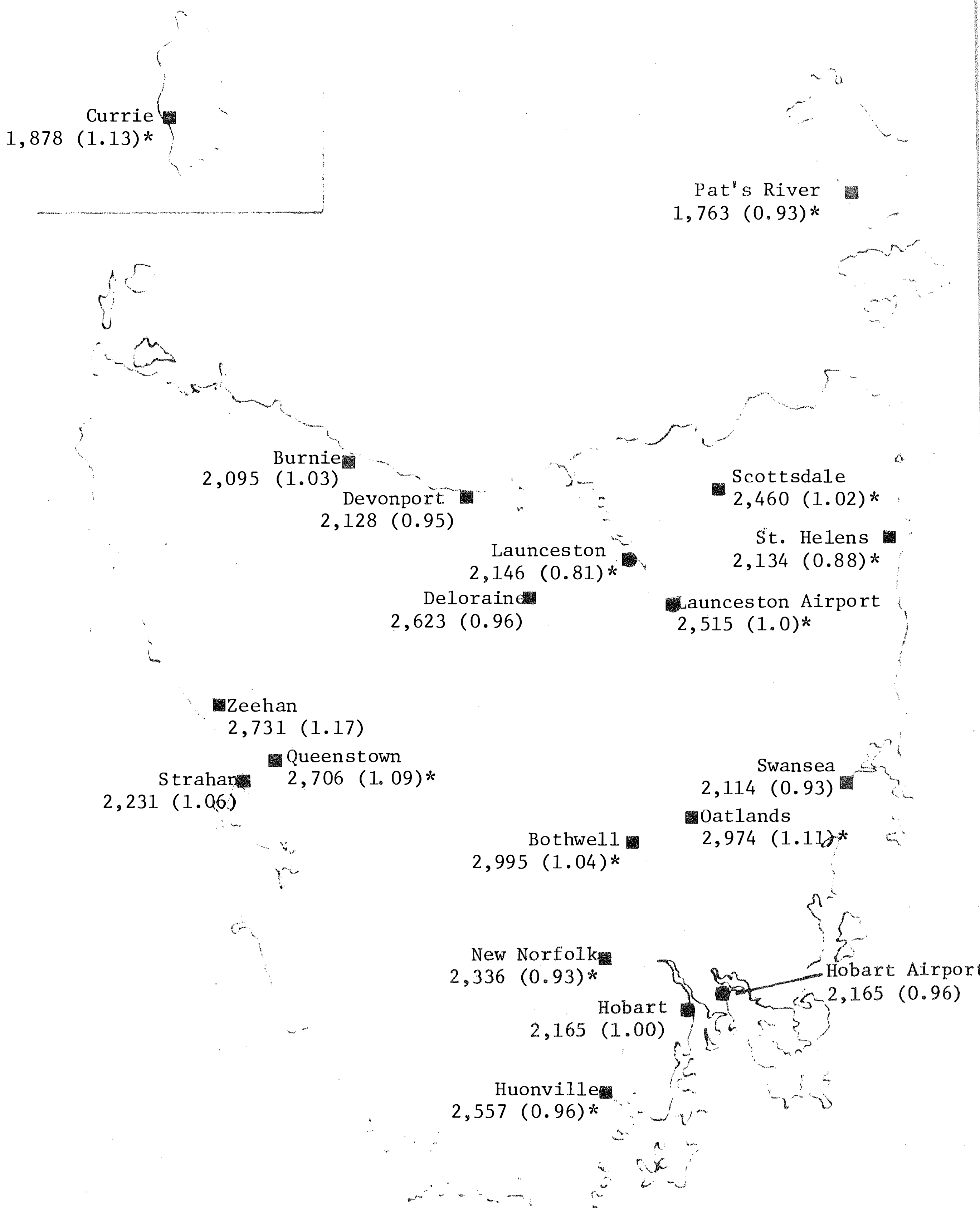
t_{min} is the mean minimum temperature for the coldest month.

The factor $(18.3 - t_{min})$ is used to account for the effects of temperature on both COP and net costs. It is unlikely that it truly represents the way in which temperatures affect these variables. However, it will give an accurate estimate when the deviations from Hobart (t_{min} = 4.4) are small. When the deviation is greater than 10 per cent ($t_m < 3^{o}$, or $t_{min} > 5.8^{o}$C) the accuracy of the estimate is questionable.

Figure 3.12 shows degree-day estimates and climatic adjustment factors for selected locations around Tasmania.

3.5 REVIEW OF COMFORT REQUIREMENTS

The temperature required by a person for thermal comfort depends primarily on his clothing and activity level. In Tasmanian homes during leisure periods, comfort requires an ambient temperature of 17-24^{o}C. At other times, comfort is provided by ambient temperatures in the range 9-20^{o}C.



* Estimates of questionable accuracy.

Figure 3.12: Heating degree-days, base 18.3°C (estimated from monthly mean temperature data) and heat pump climatic adjustment factors (in brackets) for selected locations in Tasmania

Ambient temperature is affected almost equally by air temperature (t_a) and mean radiant temperature (t_{mrt}). When t_{mrt} is low, in winter, t_a must be correspondingly higher ($\sim 2^{\circ}C$). When t_{mrt} is high (in summer, or when radiant heating is being used) t_a can be lower.

Use of the living-room heater depends on the activities engaged in by the occupants of the house. It is normally expected to provide a high standard of heating during the evening and (when the house is occupied throughout the day) at certain periods during the day. During the day, it can also be used to provide a low standard of heating throughout the house, by being used with internal doors left ajar. The two basic styles of use (evening-only and day-and-evening) will be approximated by the maintenance in the living area of an ambient temperature of $20^{\circ}C \pm 1^{\circ}C$ from 5 p.m. to 11 p.m., and from 7 a.m. to 11 p.m., respectively.

Heating loads for this type of situation are best calculated from hourly climatic data, using a computer. The loads obtained from Hobart climatic data can then be extrapolated, using degree-day data, to other locations around Tasmania.

CHAPTER FOUR: THERMAL PROPERTIES OF HOUSES

4.1 INTRODUCTION: CALCULATING HEAT LOSSES BY COMPUTER

Previous chapters have built up a picture of what the occupants of a house require for thermal comfort, and how Tasmania's climate affects the demands made on heating systems. The final factor affecting heating requirements is the house itself. The effects of insulation and different construction materials on the heating requirements of two housing division houses can be seen from the information presented in Chapter Five. Those results are the results of detailed computations using the TEMPAL computer programme developed by Alan Coldicutt at the University of Melbourne. In this chapter, we will show how these results have been derived, and evaluate the effects of two variables - house size and design - that are not adequately accounted for in the computed results.

When part of a house is maintained at a constant temperature, heat is lost by the processes of conduction, through the *building envelope* (walls, windows, floor and roof) and convection, due to loss of warm air through natural ventilation. By the same processes, some of the heat passes from the heated section of the house to the unheated section. Part of the heating load caused by these heat losses will be supplied by body heat of the occupants and heat gains from appliances and (during daylight hours) from solar radiation.

Figure 4.1 shows typical values of heat gains and losses occurring in an uninsulated house over a Hobart mid-winter's day. These values indicate the relative importance of the various paths of heat loss.

Conduction and convection heat losses, and solar radiation heat gain, vary quite markedly from house to house. For a given location and time, the radiation heat gain depends primarily on the position, size

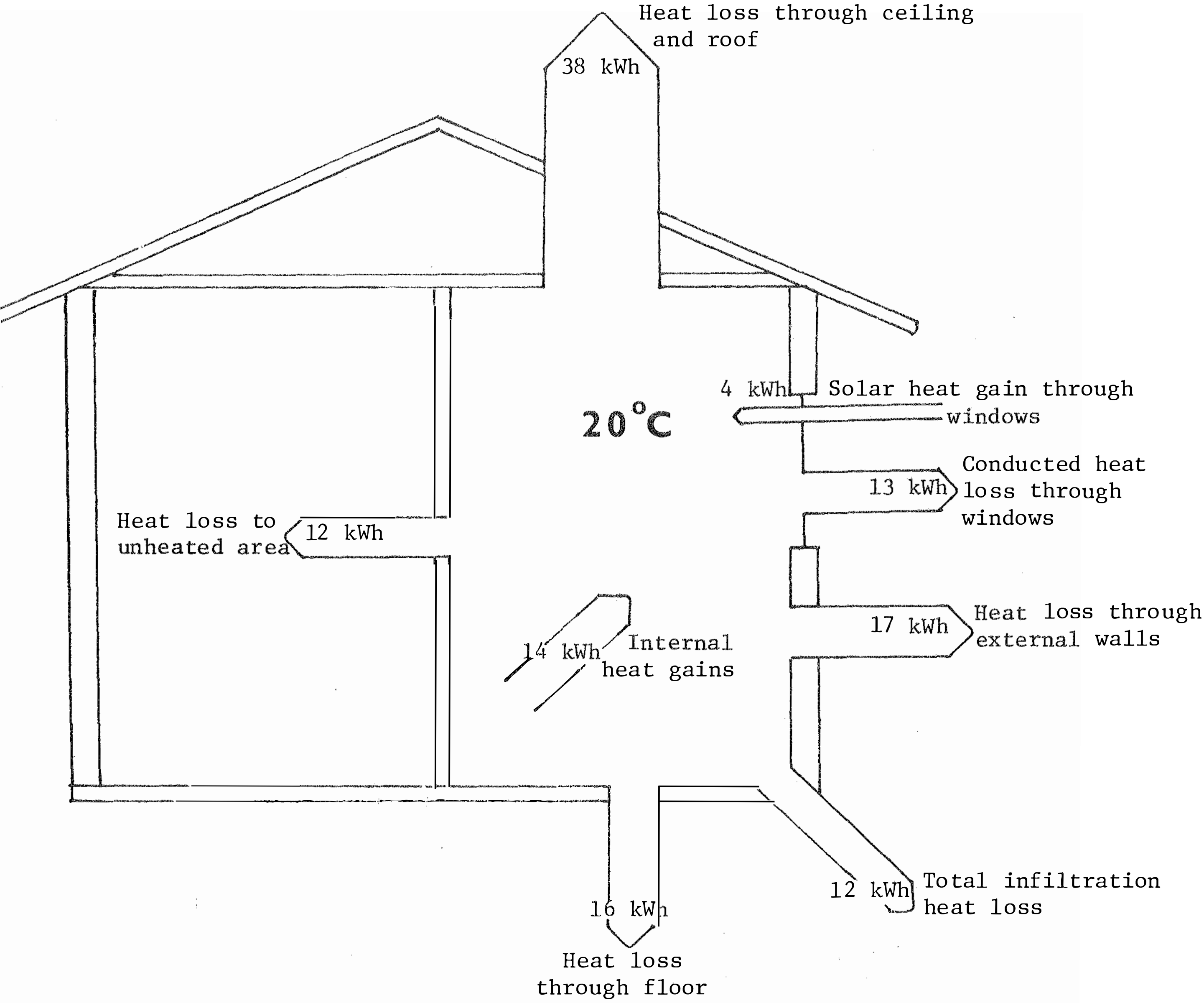


Figure 4.1: Computed heat gains and losses (excluding heater input) for an uninsulated weatherboard house, heated from 7 a.m. to 11 p.m., on a Hobart winter's day. The net heat loss of 90 kWh is close to the average daily heat loss of this house in midwinter (June and July).

and shading of windows. Convected heat losses depend on exposure to wind, and the use of measures (such as weather-stripping) to control ventilation. The major heat losses, by conduction, depend on the thermal resistance of the building envelope, which can be greatly increased by the use of insulation.

When conditions vary over time, heat storage in the building structure becomes important. Some of the heat gained from solar radiation or heating appliances serves to heat the building structure, rather than the air in the house. At night, or when the heating is turned off, the stored heat may be released, helping to maintain the indoor temperature. As a result, heavyweight buildings, with high thermal storage, do not experience the same daily extremes of temperature and heating load as lightweight buildings. This results in improved year-round comfort, though annual heating requirements are relatively unaffected (Coldicutt *et al.*, 1978). Heavyweight building materials also work in favour of the economics of heat pumps, since the lower peak heating loads can be supplied by smaller, less expensive units.

The thermal performance of a number of houses has been predicted by a computer programme (TEMPAL) which uses values of a number of thermal parameters to produce a numerical model of each house. The major aspects of this model, and the important thermal parameters, are discussed in section 4.2. With the resources available, it was possible to treat only a few out of the vast range of possible Tasmanian houses. Section 4.3 explains how the choices were made. Section 4.4 evaluates the effects of house design on heating, and the accuracy of the assumption that heating load is proportional to area.

4.2. MODELLING THERMAL PERFORMANCE

Conducted Heat Losses

The most useful measure for calculating steady-state conducted heat losses is the *U-value*. This is the amount of heat energy per second that passes through a square metre of the building envelope when a $1^{\circ}C$ difference in (air) temperature is maintained across it. The U-value is the inverse of the *thermal resistance* (R), and the heat loss per square metre of the building envelope is calculated by the equation:

$$q_e = U(t_i - t_e) \qquad (1)$$

where: q_e is the heat loss through the section of building envelope ($W.m^{-2}$).

U is the U-value of the section ($W.m^{-2}.{}^{o}C^{-1}$).

t_i is the internal air temperature (${}^{o}C$).

t_e is the external air temperature (${}^{o}C$).

To derive the U-value, one first calculates the thermal resistance of a cross-section of the building element. As an example, consider a section through a brick veneer wall (Figure 4.2). Thermal properties of the building materials can be obtained from standard texts (ASHRAE, 1972; Barned, 1970). R is calculated as the sum of all thermal resistances encountered on passing through the section, including the thermal resistances of air spaces and surfaces. The overall U-value ($\frac{1}{R}$) takes into account the effects of framing on a small proportion of wall area, and will be slightly lower than that shown in Figure 4.2. A selection of U-values is shown in Table 4.1.

It is notable that most of the thermal resistance is provided by the air space and surfaces, and only about one-third by the actual building materials. This fraction becomes even smaller when insulation is used, so that heat losses depend more on insulation and construction methods than on the actual materials of construction.

Transient effects, due to the effects of thermal storage, are treated as corrections to the steady-state heat flow. Changes in external temperature alter the amount of heat stored in the building structure. The effects of this heat filter through to internal after a time *lag* and are proportional to the *transfer modulus* of the building element. Changes in the internal temperature also affect thermal storage. In this case, the change in thermal storage occurs *after* the change in internal temperature (i.e. the change in t_i precedes thermal storage). Hence, the internal storage effect is described by a time *lead*. It is proportional to the *internal admittance* of the building element.

Values of lag, lead, internal admittance and transfer modulus are included in Table 4.1.

The heat loss equation, taking thermal storage effects into account, may be expressed as:

Heat loss = heat loss due to external storage effects
+ steady-state heat loss
+ heat loss due to internal storage effects.

In the TEMPAL programme, this is modelled by the algorithm:

$$q_e(t) = \Gamma.[\bar{t}_e(t) - t_e(t-\delta)] + U.[\bar{t}_i(t) - \bar{t}_e(t)] + \gamma.[t_i(t+\phi) - \bar{t}_i(t)] \qquad (2)$$

where:

$q_e(t)$ = heat loss through element at time t ($W.m^{-2}$)

U = overall heat transfer coefficient (steady state) ($W.m^{-2}.{}^{o}C^{-1}$)

Γ = transfer modulus (admittance) of the element ($W.m^{-2}.{}^{o}C^{-1}$)

γ = internal admittance of the element ($W.m^{-2}.{}^{o}C^{-1}$)

δ = lag (hour)

ϕ = lead (hour)

$t_e(t-\delta)$ = external temperature at time $t-\delta$ (^{o}C)

$t_i(t+\phi)$ = internal temperature at time $t+\phi$ (^{o}C)

$\bar{t}_e(t)$ = mean external temperature at time t (^{o}C)

$\bar{t}_i(t)$ = mean internal temperature at time t (^{o}C)

The mean temperatures, (external and internal) at time t, are calculated as:

$$\bar{t}_e(t) = \sum_{j=0}^{P-1} t_e(t-j)/P \qquad (3)$$

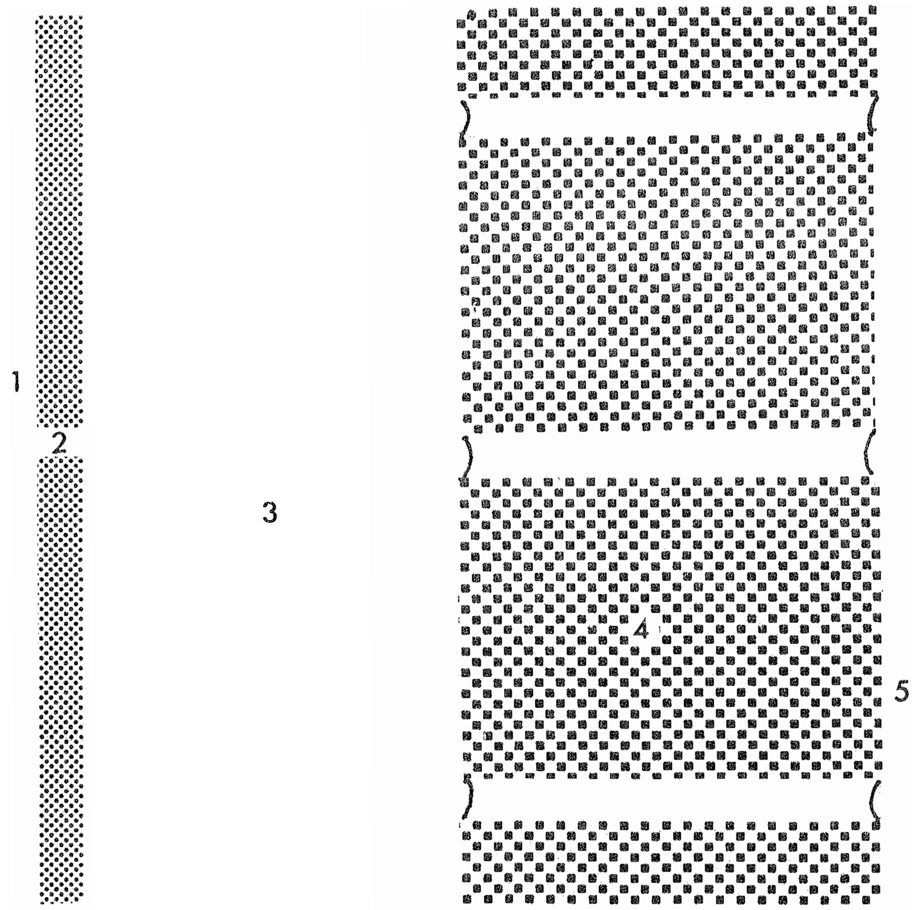
$$\bar{t}_i(t) = \sum_{j=0}^{P} t_i(t-j)/P \qquad (4)$$

where: P = response period of the element (approximated as $5.\delta$)

The mean for successive hours can be simply calculated:

$$\text{e.g.} \quad \bar{t}_e(t+1) = \bar{t}_e(t) + [te(t+1) - te(t-[P-1])]/P \qquad (5)$$

Hence, this method has been dubbed *advancing mean*. It is similar in principle to the *response factor* method (ASHRAE, 1972) but uses simpler mathematics to reduce the computation time.



			R ($m^2.{}^oC.W^{-1}$)
1:	Surface resistance (vertical surface, still air)	:	0.12
2:	12 mm. fibrous plaster	:	0.04
3:	Air space, vertical, 100 mm 10^oC	:	0.18
4:	Brick, 110 mm	:	0.10
5:	Outside surface, 25 km/h wind	:	0.03
	TOTAL THERMAL RESISTANCE		0.47

U-Value = 1/R = 2.13 $W.m^{-2}.{}^oC^{-1}$

Figure 4.2: Cross-section of a brick veneer wall, showing thermal resistances (no correction made for thermal resistance of framing)

Table 4.1: Selected thermal properties of building elements.

Building element	Direction of heat flow	Insulation	Materials	U-value $W.m^{-2}.^{o}C^{-1}$	Transfer modulus $W.m^{-2}.^{o}C^{-1}$	Lag(δ) hours	Internal admittance $W.m^{-2}.^{o}C^{-1}$	Lead(ϕ) hours
Roof/ ceiling	up	nil	Metal deck/plasterboard	2.60	2.59	0.3	2.65	0.5
			T.C. tiles/plasterboard	2.47	2.46	0.6	2.54	0.6
		RFL sarking (dusty surface)	Metal deck/plasterboard	1.17	1.16	0.4	1.38	1.8
			T.C. tiles/plasterboard	1.07	1.07	0.7	1.32	2.0
		75 mm mineral wool batts	Metal deck/plasterboard	0.38				
			T.C. tiles/plasterboard	0.38	0.38	1.2	1.00	4.0
		RFL + 75 mm batts	Metal deck/plasterboard	0.32				
			T.C. tiles/plasterboard	0.32	0.32	1.3	1.00	4.3
External wall	in/out	nil	Asbestos cement/plasterboard	2.39	2.38	0.4	2.44	0.6
			Timber/plasterboard	1.82	1.80	0.8	1.95	1.1
			Brick veneer/plasterboard	1.98	1.63	3.1	2.33	1.1
			100 mm concrete block veneer	1.94	1.53	3.4	2.34	1.2
			Brick cavity/plasterboard	1.77	0.87	6.8	4.78	1.6
			Solid concrete 150 mm	3.51	2.50	3.9	5.23	1.2
		Urea formaldehyde foam (40 mm)	Cavity brick	0.64	0.29	7.3	5.41	2.1
		RFL (dished)	Brick veneer	0.64	0.50	3.6	1.07	3.0
			Weatherboard	0.63	0.62	1.0	1.04	3.1
		50 mm batts	Brick veneer	0.51	0.39	3.8	1.03	3.5
Windows		nil	No curtains	6.00				
			Curtains drawn	4.80				
Internal wall	horizontal	nil	Plasterboard on studs	1.80	1.78	0.8	1.94	1.1
Floor	down	nil	Timber (suspended)	2.16	2.14	0.6	2.22	0.6
			Timber and carpet	1.65				
			100 mm concrete (suspended)	2.56	1.56	4.1	4.59	1.2
			125 mm suspended concrete slab with wood block or heavy carpet	1.82	0.75	5.9	2.97	0.8
			Concrete slab on ground	0.60	0.00	14.4	4.62	0.8
			Concrete slab on ground and carpet	0.50	0.00	15.0	2.90	0.6
		RFL	Timber	0.78	0.76	0.9	1.29	2.7

Table 4.2: Air changes taking place under average conditions, exclusive of air provided for ventilation[1] (ASHRAE, 1972).

Kind of room or building	Number of air changes taking place per hour
Rooms with no windows or exterior doors	½
Rooms with windows or exterior doors on one side	1
Rooms with windows or exterior doors on two sides	1½
Rooms with windows or exterior doors on three sides	2
Entrance halls	2

[1] For rooms with weather-stripped windows or with storm sash, reduce rates to two-thirds of the values shown.

Ventilation (infiltration) Heat Losses

Ventilation rates are commonly expressed in terms of the number of air changes per hour. The ventilation heat loss is equivalent to the heat required to raise the new air to the internal temperature, and may be calculated from the equation (ASHRAE, 1972):

$$q_v = 0.335 \cdot c \cdot V \cdot (t_i - t_e) \qquad (6)$$

where:
- q_v = ventilation heat loss (Watts)
- c = number of air changes per hour (hr^{-1})
- V = volume of heated space (m^3)
- t_i = internal air temperature (^oC)
- t_e = external air temperature (^oC)

General air change rates can be estimated from Table 4.2. The air change rate for the whole house is normally approximated as the sum of the rooms. Coldicutt *et al.* (1978) have related air change rates to wind velocities with simple formulae derived from the results of monitoring specific houses. For the brick veneer house (type 305B), the air change rate is approximated by the expression:

$$c = 0.5 + 0.2\,v \qquad (7)$$

where: v = wind velocity ($m.s^{-1}$)

Wind velocities are generally lower for houses than for exposed meteorological stations. Wind velocities for houses are estimated by multiplying the recorded value by a wind factor: 0.8 for exposed sites, and 0.5 for sheltered sites. The air change rate (equation 7) can be used to derive the rate of heat loss (equation 6) through infiltration.

It has been assumed that external doors and doors between zones remain shut. If the doors between the heated and unheated zones remain open, the annual heating energy can be twice as great as that required to heat only the living zone (Coldicutt *et al.*, 1978). Occasional use of doors will have little effect on heating loads.

Internal Heat Gains

Occupancy

A normal-sized person, seated quietly, produces approximately 100 watts of heat from metabolic processes. This rate doubles if the person becomes reasonably active (e.g. moderate housework), and falls to 70 watts during sleep (refer to Chapter Three for details).

Lights and Appliances

Cooking appliances produce heat directly. Other appliances may produce mechanical energy, which is mostly converted to heat through friction. Incandescent lighting converts only 7 per cent of its electrical energy into light energy, and loses the rest directly as heat. Fluorescent lights are somewhat more efficient (20 per cent) but, in any case, the light is mostly absorbed by objects within the room and converted to low-grade heat. Thus, virtually all the electrical energy used in a house contributes toward the heating load.

Some appliances, such as washing machines and clothes dryers, are normally located outside the heated area, and are not expected to affect heating.

A double electric blanket, food blender, floor polisher, hair dryer, iron, kettle, food mixer, radio, cooking range, refrigerator (two-door automatic defrost), television (black and white) and vacuum cleaner, with average use result in an electricity bill of $113.50 per year (Hydro-Electric Commission News, October-December 1977). At 3.42 c/kWh, this is an average of 9.1 kWh per day.

Lighting has been estimated, for normal usage, at 1.9 kWh per day.

The contribution from hot water is more difficult to determine. Much of the hot water is used outside the heated area, and part of its heat goes literally down the drain without contributing to heating. Per capita hot water energy use is 3.978 GJ per annum (Hartley, Jones and Badcock, 1978). At an average occupancy rate of 3.3 persons per dwelling, the daily rate is 10 kWh. Assuming that 7 per cent of this heat contributes to heating, the hot water component becomes 0.7 kWh per day.

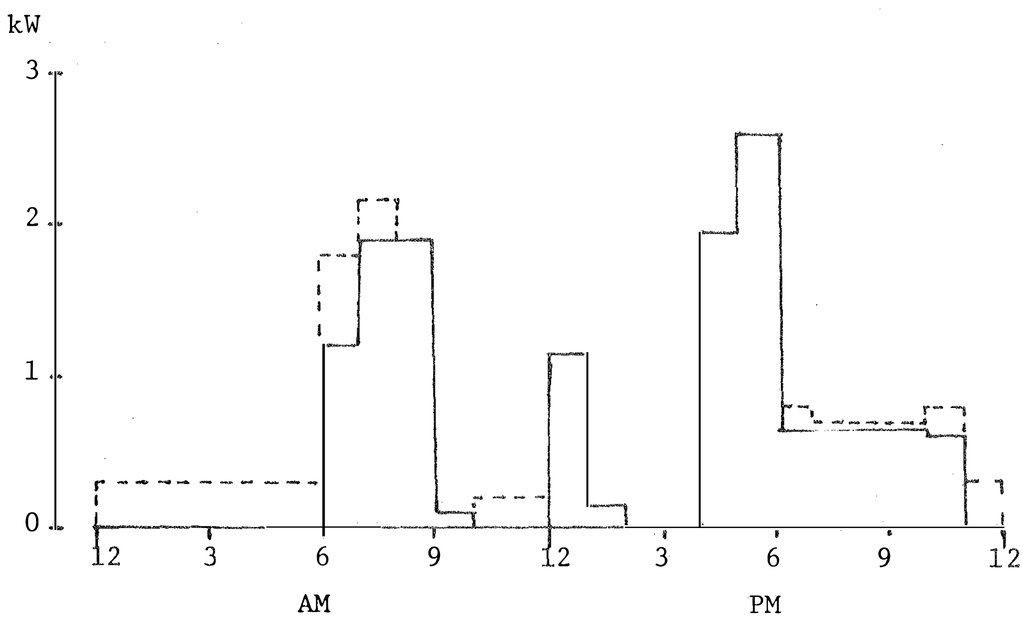
Estimated hourly incidental and occupancy heat gains are depicted in Figure 4.3.

Solar Radiation Heat Gain Through Windows

Architectural glass is transparent to the short-wavelength radiation emitted at high temperatures by the sun, but opaque to the long-wavelength radiation emitted by objects at room temperature. Thanks to this "greenhouse effect", the only significant amounts of heat radiated through windows come from sunlight and help to reduce the winter heating load.

Solar radiation incident on a window may be transmitted, reflected or absorbed. Absorbed radiation serves to raise the temperature of the glass, and is eventually passed to the indoor or outdoor air as convected heat. The ratio of transmitted to reflected radiation varies with the angle at which the radiation strikes the glass. Typically, 85-90 per cent of the incident radiation is transmitted. When a window directly faces the sun, virtually no radiation is reflected. As the angle between the sun and the window increases, the proportion of transmitted light decreases. So also does the area presented by the window normal to the sun.

Clear-sky solar heat gains can be calculated from geometrical considerations, for windows of any given orientation and shading, and for any given latitude and time (e.g. Spencer, 1977). In the TEMPAL computer programme, the *clear-sky* radiation gains are calculated for each hour of the median day of each fortnight, in order to eliminate unnecessary computer time. Actual solar radiation heat gains are



Key:

Heated area: ————

Whole house: - - - -

Figure 4.3: Assumed daily pattern of incidental and occupancy heat gains

calculated by simple factoring, using recorded values of hourly solar radiation intensity.

For calculating conducted heat losses, windows are assumed to be bare ($U = 6.0$) during the day, to gain solar heat, but curtained ($U = 4.8$) at night (as assumed by Coldicutt *et al.*, (1978).

Solar Radiation Heat Gain Through Opaque Surfaces

When sunlight falls on the surface of a wall or roof, a fraction of the radiation is absorbed, causing a rise in the surface temperature. This rise in temperature will increase the heat lost from the surface by radiation and convection, and decrease the temperature gradient across the building element. [For a more complete description, see ASHRAE (1972).] The increased radiation and convection heat losses do not directly affect internal temperatures, but the decreased thermal gradient reduces heat loss from indoors.

It has been found that this effect can be adequately modelled by the use of a *sol-air temperature*, defined as the (fictitious) outdoor air temperature at which the same rate of heat loss from the surface would exist, in the absence of all radiation exchanges. The sol-air temperature, t_{sa}, is used in equation (2) in place of the external temperature, t_e.

An expression for t_{sa} is given by Walsh (1977):

$$t_{sa} = t_e + (\alpha I - \phi T_e^3)/h$$

where:

$$\phi = \begin{cases} 4\,\sigma\varepsilon\,(14 - 0.2\,t_e)\,(1 - m/9) & \text{for horizontal surfaces} \\ 8\,\sigma\varepsilon\,(5 - 0.1\,t_e)\,\arctan\,[100/r]\,(1 - m/9)/\Pi & \text{for vertical surfaces.} \end{cases}$$

t_{sa} = sol-air temperature (^oC)

t_e = outdoor air temperature (^oC)

T_e = outdoor air temperature (K)

= $t_e + 273.15^oC$

α = solar absorptance of the surface

I = total (global) solar irradiance incident on the surface ($W.m^{-2}$)

h = $11.3 + 3.31\,v$ ($m.s^{-1}$)

v = wind speed ($m.s^{-1}$)

σ = Stefan - Boltzmann constant

ε = emittance of surface to long-wave radiation

m = sky cloudiness number or total cloud cover (oktas).

Walsh (1977) has calculated that in a house of low thermal resistance, solar heat gain by conduction through the building envelope can be greater than the gain by transmission through windows. So the total effect of conducted solar radiation heat gain, as modelled by t_{sa}, is quite important.

When comparing different house designs, with different window areas, differences in the radiation heat gain through opaque surfaces will be negligible provided that the differences in window area represent only a very small proportion of the total opaque area.

The TEMPAL Computer Programme

The nature of the simulation can be followed by tracing the use of the programme via the flow chart (Figure 4.4).

The user calls up TEMPALDATA and inserts all data other than climatic data at the terminal. When run, this programme generates files D1, C1 and A1. Permanent files D3 and D4 are generated from weather tapes supplied by the CSIRO Division of Building Research. D3 contains hourly data of solar radiation (normal to the beam, and diffuse on a horizontal surface) and wind speed. D4 has hourly data of air temperature (dry bulb), cloud cover and relative humidity.

SOLINFO uses information from local file D1 to calculate hourly values of *clear sky* solar radiation transmitted through windows and absorbed by other building components, for the *median day* of each fortnightly period. SOLCON combines the output from S1 with actual figures for hourly solar radiation and wind speed from D3. By simple factoring, it calculates actual absorbed and transmitted solar radiation.

SHADPROG uses data from S2 and data input on X,Y,Z co-ordinates of all windows and opaque elements, and external objects, to calculate "effective" areas for each element for diffuse radiation and hourly areas of each element in sun. CLIMCON uses the output from SHADPROG and data from local files S4, S5, S6 and D4. Output from CLIMCON is written on local files S7 (air dry bulb temperatures, actual solar

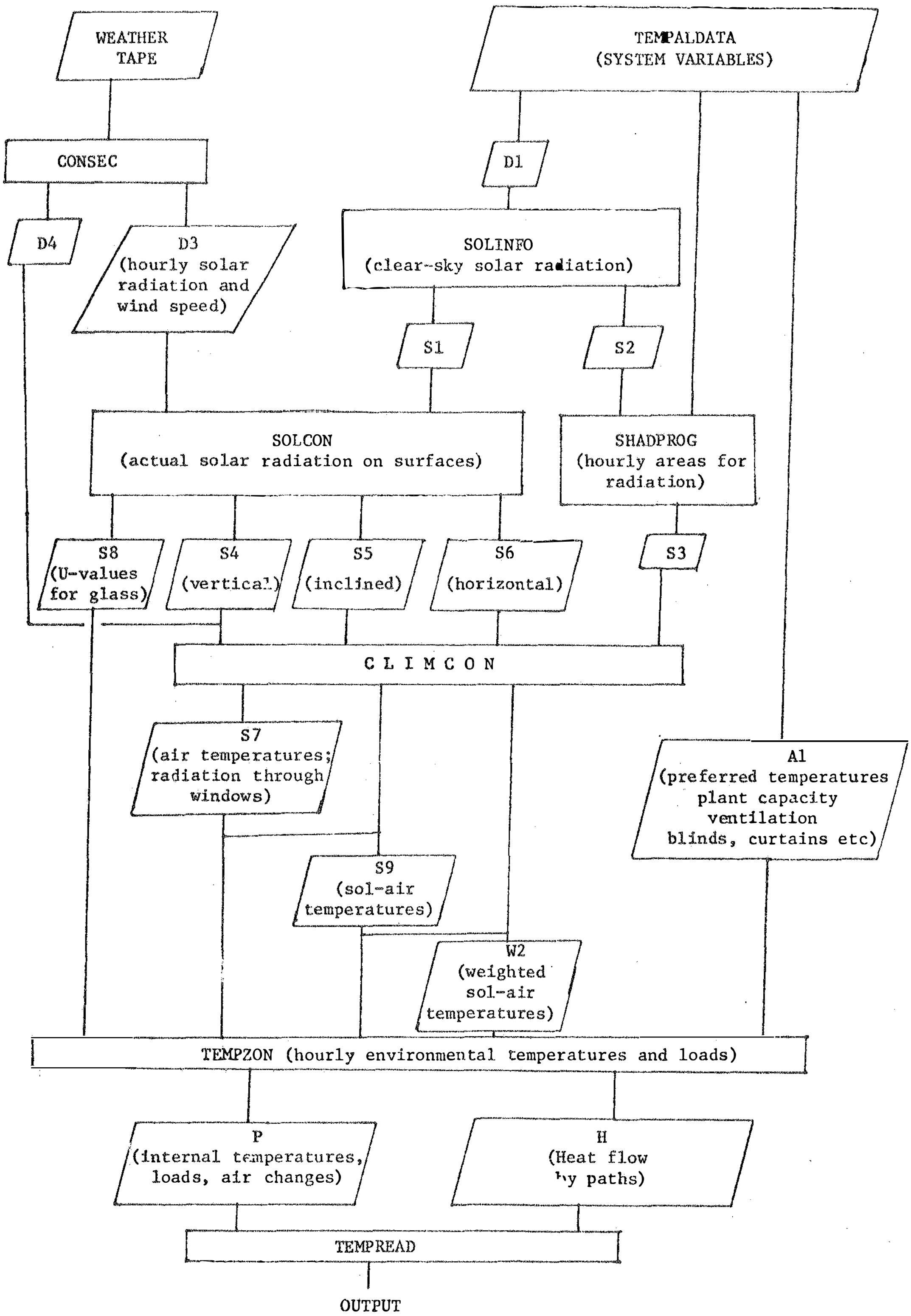


Figure 4.4: TEMPAL Flow Diagram

radiation through windows), S9 (sol-air temperatures of all surfaces) and W2 (weighted sol-air temperatures of walls).

TEMPZON is the central programme. At this stage, division of the elements into the respective zones is determined. Information related to user requirements (preferred temperatures, etc.) is read from local file A1 and combined with that from S7, S8, S9 and W2 to predict hourly environmental temperatures and loads. Information for each hour of the run is written on files P (internal temperatures, loads and air changes) and H (heat flow by path analysis).

TEMPREAD reads local files P and H and prints out an analysis of energy totals (daily and cumulative), frequency analyses of loads and environmental temperatures, hourly information of these for nominated days and the analysis of heat flow by paths.

The accuracy of the TEMPAL programme has been demonstrated by a comparison of actual and predicted temperatures inside two Tasmanian housing division houses at Bridgewater, which were each monitored over a period of one week in October, 1977.

The results of the monitoring are shown in Figures 4.5 and 4.6.

4.3 SELECTION OF HOUSES FOR COMPUTER MODELLING

House Design

The two designs selected – Tasmanian Housing Division types 305 and 314 (Figures 4.7 and 4.8) - have already been selected for evaluation of their thermal performance (Coldicutt *et al.*, 1978). New computations were required to allow heat pump COP's (which depend on the outdoor air temperature associated with the heating load) to be determined. The new computations also assume a different living zone temperature (20°C - refer to Chapter Three) and include houses of weatherboard construction.

Detached houses were chosen, as they are the type of dwelling traditionally preferred in Tasmania and still comprise three out of four new dwelling units (*Tasmanian Year Book*, 1978). Other types of dwellings, because of wide variations of design, could not be covered adequately in a report such as this. House type 305 is currently built by the Housing Division in both brick veneer and concrete block veneer.

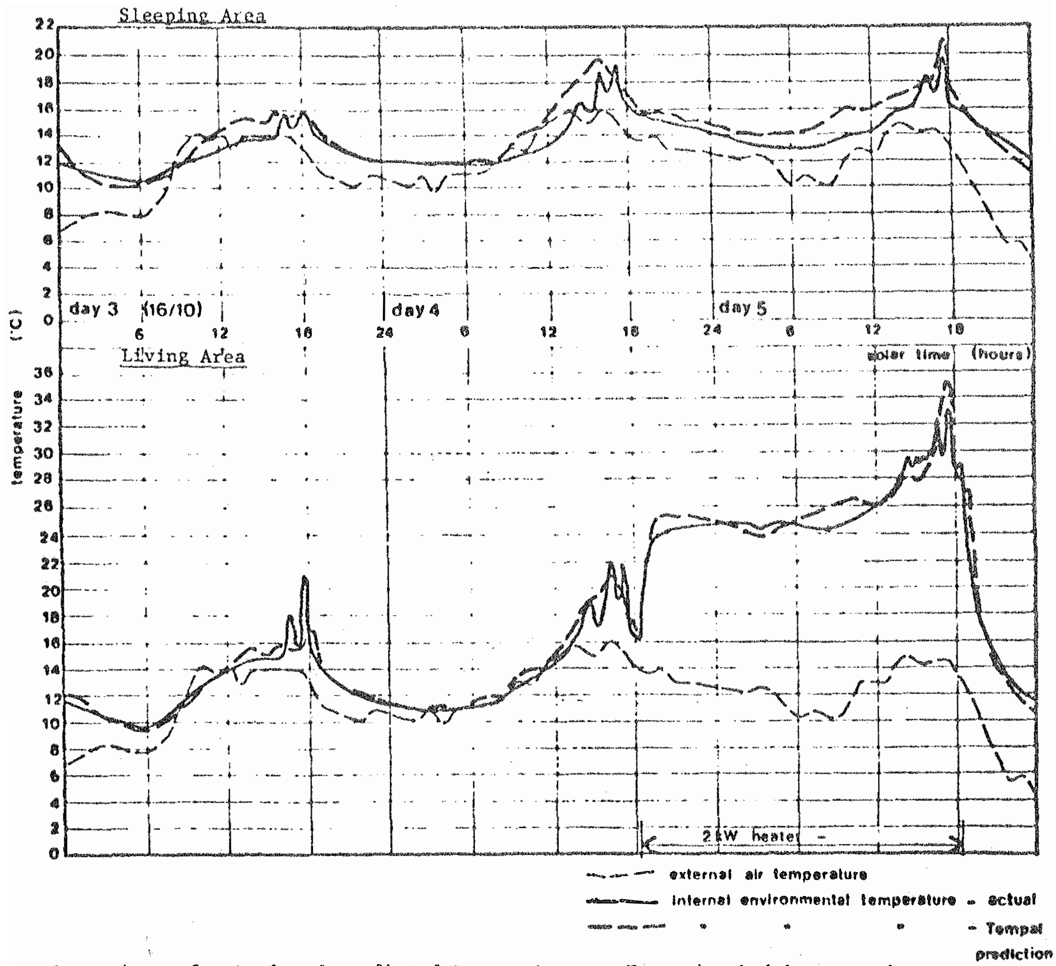


Fig. 4.5 Comparison of actual and predicted temperatures - Tasmanian brick veneer house
NOTE: TEMPAL predictions are average hourly temperatures.

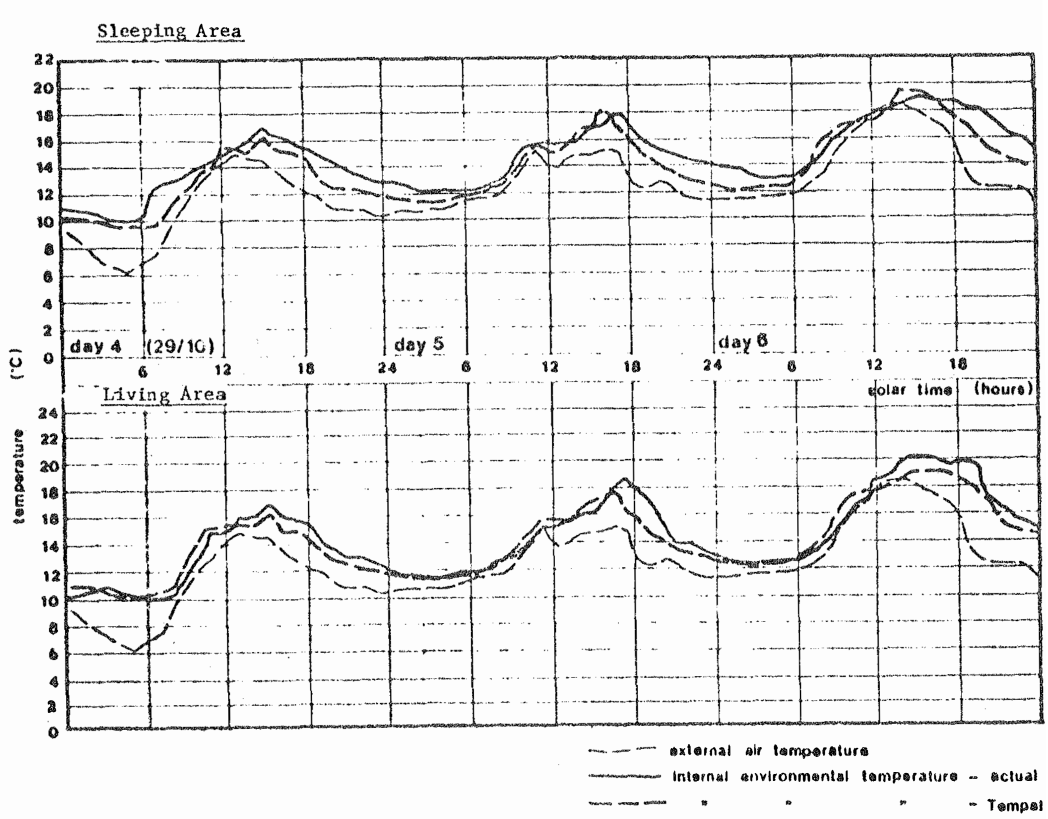


Fig. 4.6 Comparison of actual and predicted temperatures - Tasmanian concrete block house
NOTE: TEMPAL predictions are average hourly temperatures.

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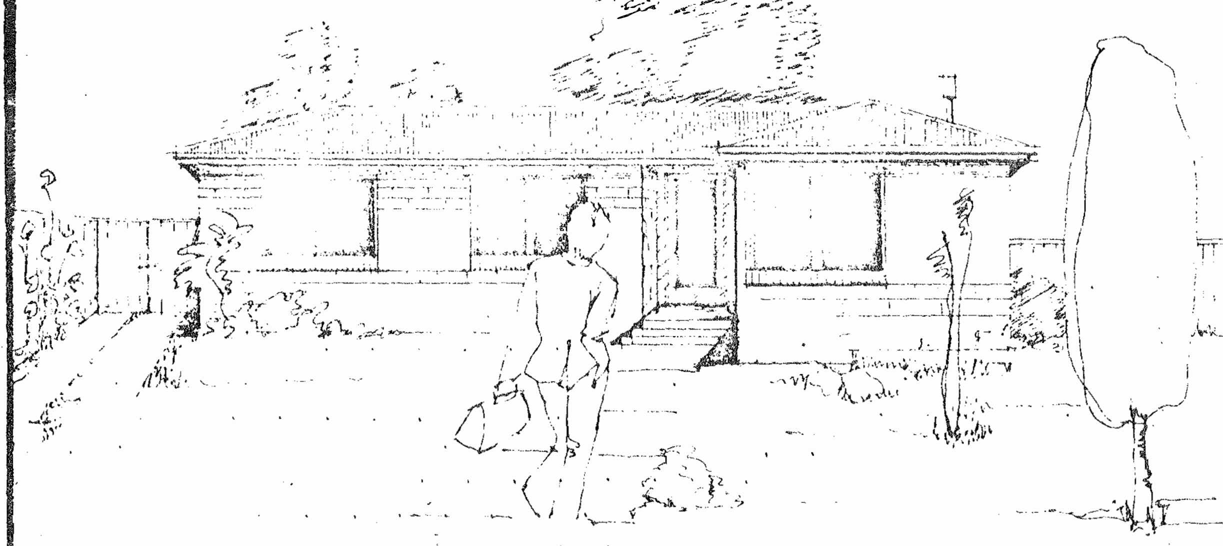
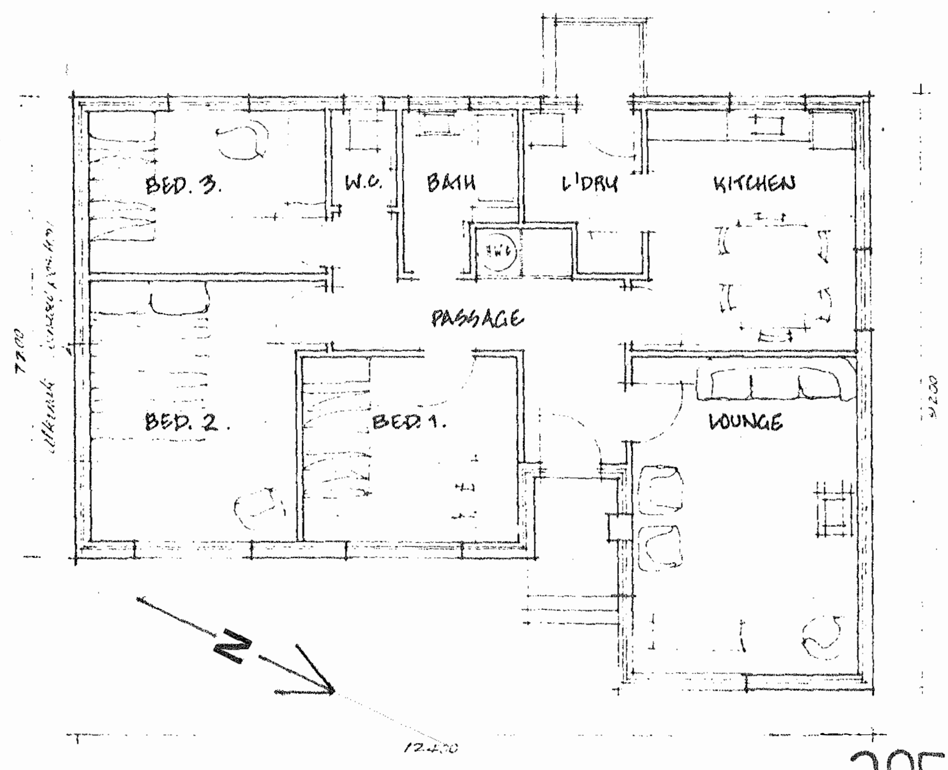


AREAS : m²

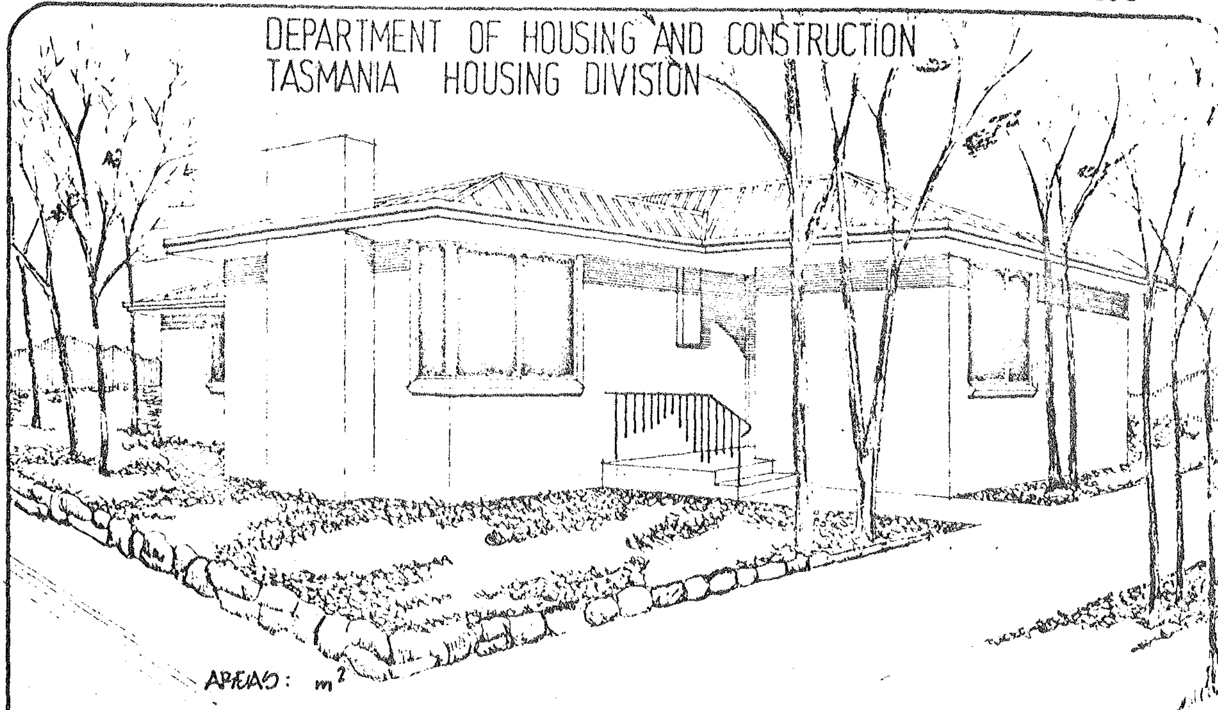
LIVING ROOM	17.52	W.C.	1.52
KITCHEN	12.49	LAUNDRY	3.93
BEDROOM 1	9.80	CIRCULATION	11.41
BEDROOM 2	13.95		
BEDROOM 3	9.48	NETT AREA	84.03
BATHROOM	3.93	GROSS AREA	97.22



305

Figure 4.7

Note: the house design shown here is a mirror image of the one used in the computer simulations.



AREAS: m^2

LOUNGE	18.87	W.C.	1.50
KITCHEN	17.87	LAUNDRY	4.15
BEDROOM 1.	13.78	CIRCULATION	12.53
BEDROOM 2.	9.48		
BEDROOM 3.	9.77	NET AREA	92.07
BATHROOM	4.34	GROSS AREA	104.14.

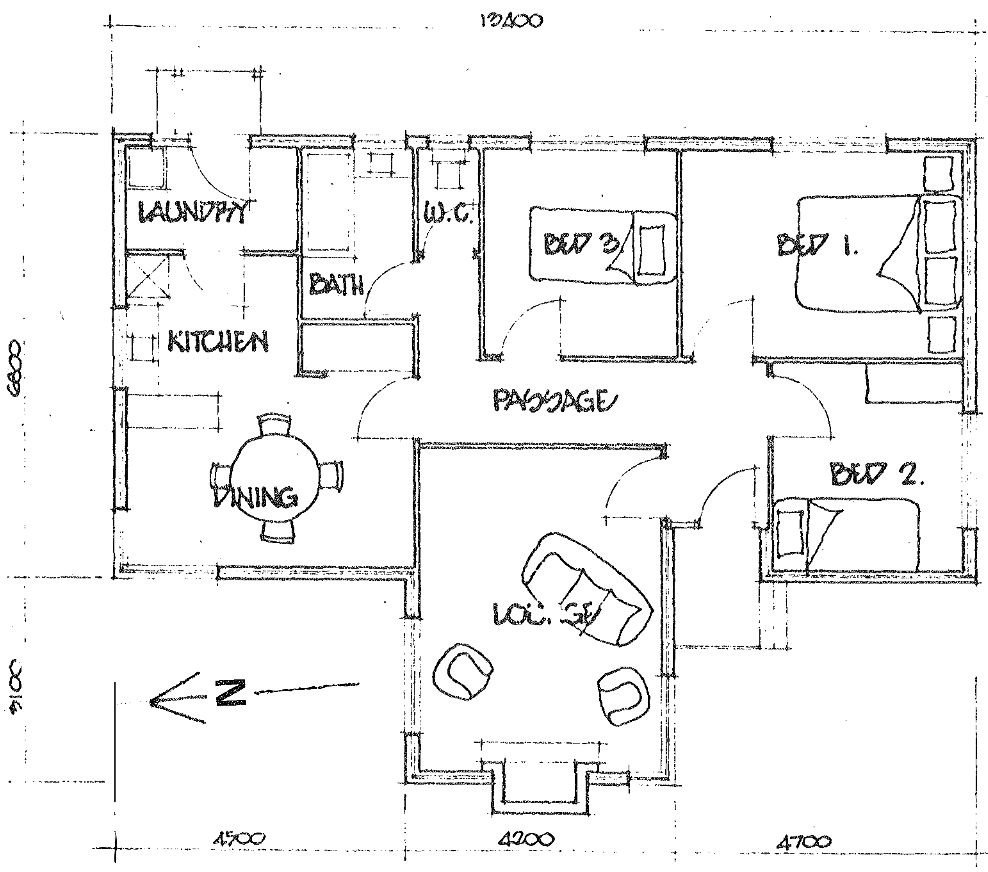


Figure 4.8

House type 314 is currently built in concrete block veneer, and was built in weatherboard until that form of construction was discontinued by the Division.

Two other designs - McArthur LH and Mayfair LH (Figures 4.9 and 4.10) - were chosen from those offered by a major private builder (Jennings Industries Ltd.). These houses have not been used in the computer simulations, but their important thermal characteristics will be compared in section 4.4 with those of the two Housing Division designs.

Ceiling and Roof

Most public housing currently being constructed has galvanised corrugated iron roofing pitched at 15^{o}, with reflective foil laminate (RFL) sarking and nominal 75 mm loose-fill fibreglass insulation. Measurements have shown that the actual amount of insulation installed is 100 mm, equivalent to 75 mm mineral wool batts. The resultant U-value is $0.32\ W.m^{-2}.^{o}C^{-1}$.

Privately-built homes commonly have terra cotta tile roofing, pitched high enough (24^{o}) to allow sarking to be dispensed with. Ceiling insulation is usually considered as an option to be retrofitted at a later date. Without RFL sarking, the addition of 75 mm mineral wool batts would bring the U-value down from $2.47\ W.m^{-2}.^{o}C$ to $0.38\ W.m^{-2}.^{o}C$.

The thermal performances of ribbed metal deck and corrugated asbestos sheeting are expected to be similar to that of corrugated iron. Two levels of insulation are considered: none, and 75 mm mineral wool batts plus RFL sarking. Although the Fibreglass Insulation Manufacturers' Association of Australia recommends the use of 100 mm batts in Hobart, the cheaper 50 mm or 75 mm batts are often preferred. The two standards chosen represent the lowest and highest U-values expected. An uninsulated tile roof will perform slightly better than the uninsulated corrugated iron roof, and with 75 mm insulation will perform almost as well as the corrugated iron roof with RFL and 75 mm batts.

Walls

Construction statistics (Table 4.3) show that over 85 per cent of houses are built of either weatherboard or brick veneer. Whilst

Table 4.3: Materials of construction (outer walls) of Tasmanian houses.

Materials	Existing occupied houses 1971 (%)[1]	Houses completed during period 1964-1975 (%)[2]	1975-1976 (%)[3]
Brick - solid	29.5	4.72	5.1
- veneer		73.40	81.9
Wood	62.6	14.5	4.0
Fibro-cement	3.5	4.9	6.3
Other	4.5	2.4	2.7

1 *Tasmanian Year Book*, 1976

2 *Tasmanian Year Book*, 1977

3 *Tasmanian Year Book*, 1978

Table 4.4: "Standard" Tasmanian house, and variations considered.

Building element	"Standard" house	Variations considered	Variations with similar effects
Design	305	314, McArthur, Mayfair	
Ceiling/ roof	corrugated iron + RFL + 100 mm loose-fill fibreglass insulation	tile roof	asbestos, metal deck tile, corrugated iron with 75 mm batts
		uninsulated c.g.i. roof	uninsulated tile, metal deck, asbestos
Walls (exterior)	uninsulated brick veneer	block veneer	asbestos cement
		weatherboard	cavity brick
		d.s. RFL insulation	block veneer, weatherboard, asbestos + RFL or 50 mm batts; cavity brick and urea foam
Floor	suspended timber, partially carpeted		suspended concrete

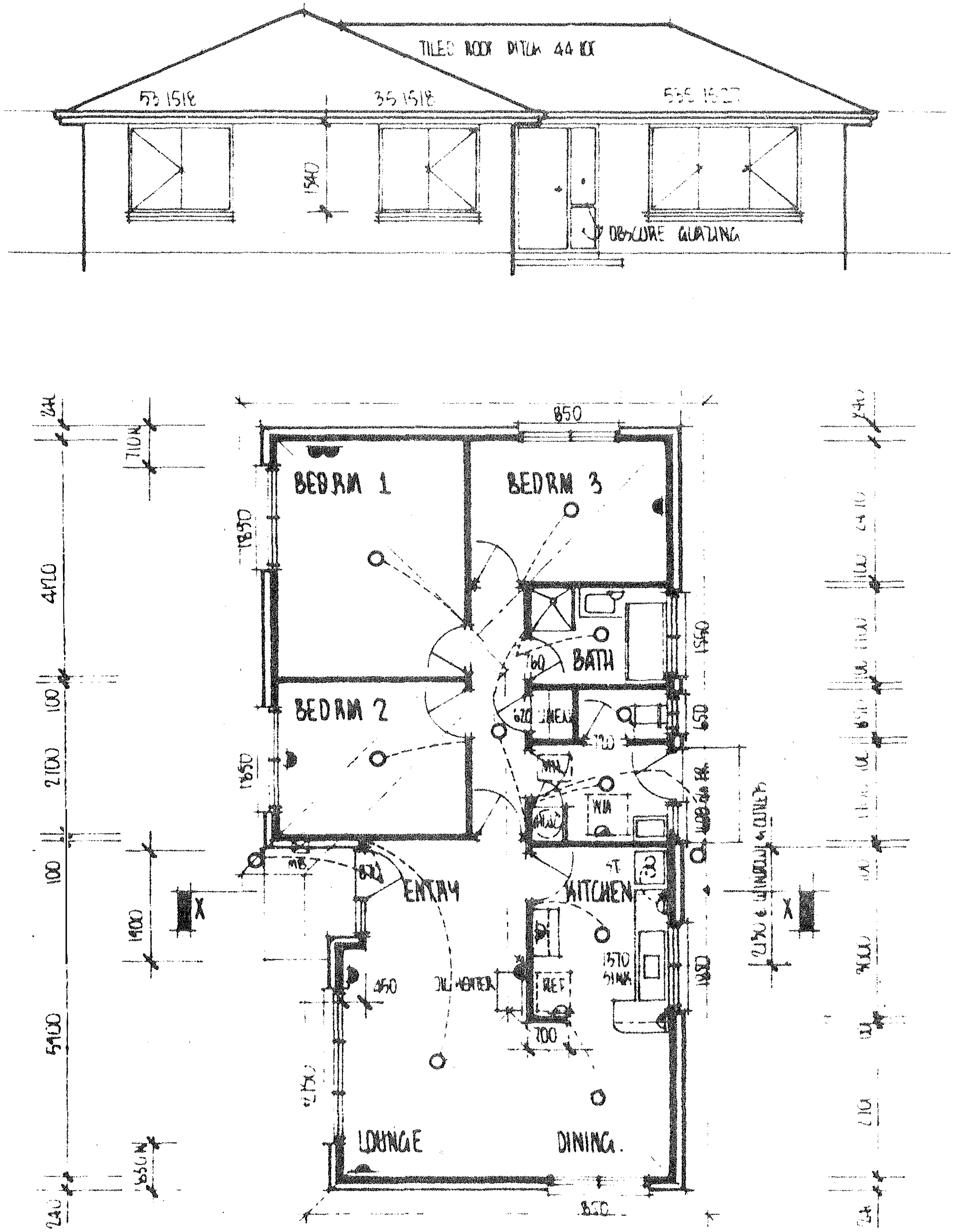


Figure 4.9: Jennings House, McArthur

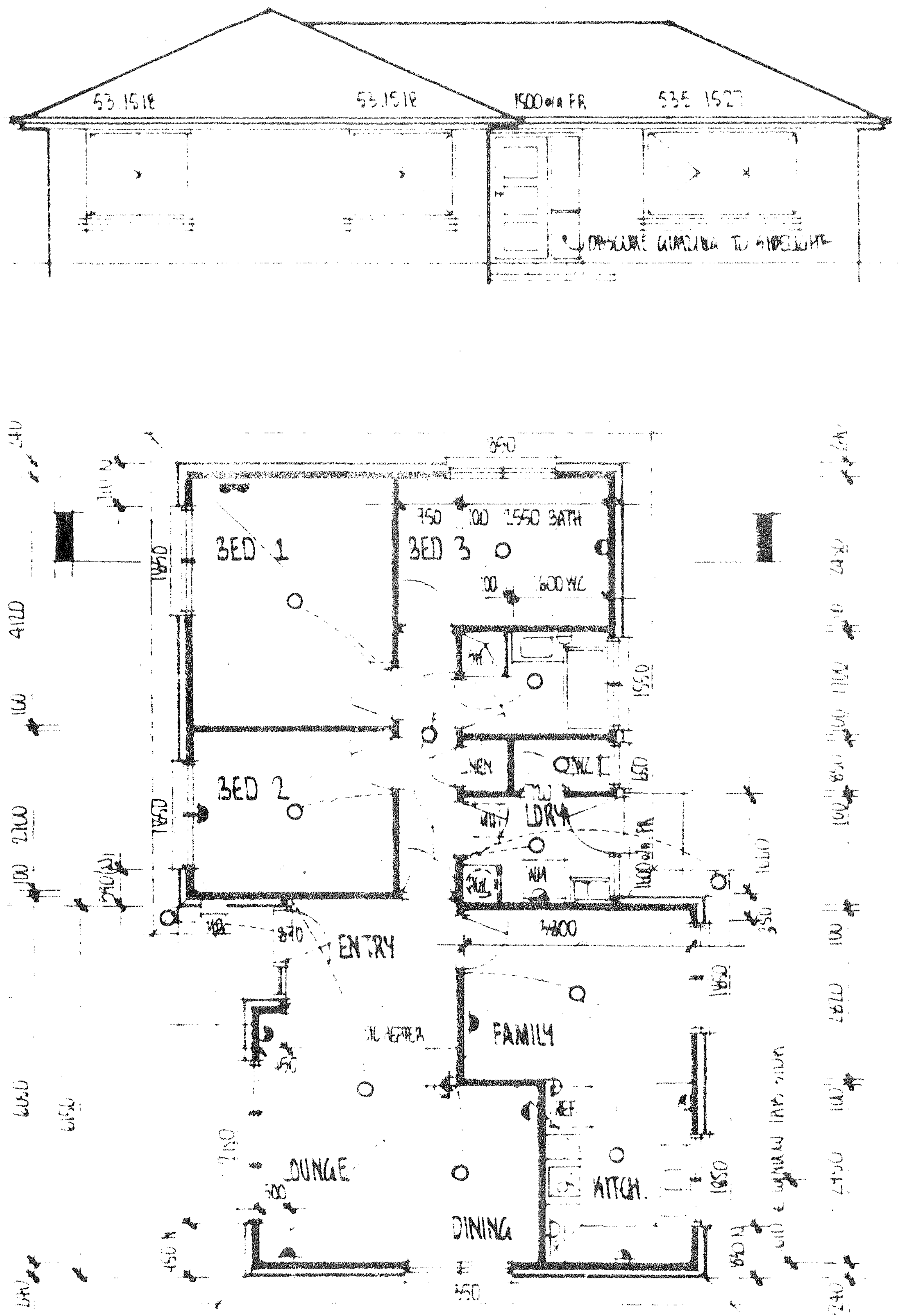


Figure 4.10: Jennings House, Mayfair

weatherboard has a slightly better thermal resistance (Table 4.1), brick veneer has a slightly longer thermal response period. Fibro-cement can be expected to perform somewhat worse than either of these, whilst cavity brick construction has a slight superiority in both U-value and thermal response period. Uninsulated solid walls (e.g. solid concrete, stone) have poor thermal resistance, and will require considerably larger amounts of heating.

The most cost-effective insulation, for the weatherboard and brick veneer walls being evaluated, is double-sided RFL (reflective foil laminate). Best performance is obtained by having the RFL dished between the studs, so that there is an air space on either side. The resultant U-value is almost identical to that obtained by filling a cavity brick wall with urea foam.

Doubts have been expressed about the possibility of condensation problems associated with the use of RFL in walls. Where this is likely, bulk insulation may be used. 50 mm batts give a U-value slightly better than that obtained by the use of RFL.

Unlike Britain, which requires a U-value of 1.0 or less, Australia has no thermal insulation standard for homes, so that insulation is at the owner's option. Wall insulation is normally considered as of secondary importance to roof insulation.

Internal walls are assumed to be plasterboard on studs. Solid brick walls may be preferred in some cases,to increase heat storage, but the thermal properties of internal walls have little effect on heating economics, since heat that passes through them is not "lost".

Floors

Floors are traditionally made of timber, on a timber frame. Private builders offer the choice of concrete slab floors (at no extra cost on level sites only). Suspended concrete floors have higher U-values than timber floors, whilst concrete slab-on-ground floors have very low U-values and long thermal response times.

Only suspended timber floors have been considered. Heating demand is expected to be slightly lower for concrete slab-on-ground floors, and slightly higher for suspended concrete floors. All floors are assumed carpeted, except in the kitchen, bathroom, laundry and toilet.

Floor insulation is uncommon unless in-floor heating is used. RFL can decrease the U-value of a wooden floor to 0.78, but becomes less effective as dust accumulates on its surface. This U-value is slightly higher than that (0.6) of concrete slab-on-ground.

The variations are summarised in Table 4.4.

4.4 HEATING REQUIREMENTS AND HOUSE DESIGN

House size is normally measured in terms of floor area, and manufacturers of heating appliances often refer to the *area* (usually of a room with an insulated ceiling) that can be heated by the appliance. In Tasmania, the figure used is about 7.5 m^2 per kilowatt. Annual heating loads, too, can be scaled according to area heated, so that the results obtained in this study can be applied to other houses, larger or smaller than the two Housing Division designs.

For most purposes, it can be assumed that both heating capacity and annual energy requirements are directly proportional to area.

Some factors which affect this relationship (such as insulation and climate) are discussed elsewhere in the report. Two sources of error will be considered here. The first, heat losses not directly proportional to area, can be accurately allowed for by a more complex relationship. The second, due to differences of design, cannot be fully accounted for without specific reference to design of the house in question.

Heating Requirements and House Size

When a house design is changed in size, retaining the same general shape, the ceiling height normally remains constant at 2.44 metres. This means that areas of walls and windows change in *direct* proportion to the linear dimensions, whilst floor and ceiling areas, and the enclosed volume, vary in proportion to the *squares* of the linear dimensions.

Thus, infiltration heat losses, and heat losses by conduction through the floor and ceiling, do in fact vary in direct proportion to the floor area of the house.

Heat gains and losses through windows and walls, however, vary in proportion to the square root of the floor area.

To provide an example of the relationship, the heat gains and losses of a concrete block veneer house, according to path, were computed for a winter's day. The house (Housing Division type 314) has its ceiling insulated with RFL and 100 mm of loose-fill fibreglass insulation. Net gains and losses are shown in Table 4.5.

Assuming that internal heat gains are independent of house size, the following "ideal" relationship is derived:

$$q_h = 0.66\ A + 5.56\ (A)^{\frac{1}{2}} - 14.4$$

where: q_h = heating deficit to be supplied by heater (kWh)
A = area of heated space (m^2).

This is compared, in Figure 4.11, with the relationship normally assumed (in this case, $q_h = 1.16\ A$). The assumed relationship (heat requirement $\propto$ floor area of heated space) underestimates heating requirements by up to 3 per cent for smaller areas, and overestimates them by up to 8 per cent for larger areas (100 m^2).

Heating Requirements and House Design

The major difference between public and private housing in Tasmania is the use of ceiling insulation, which reduces the ceiling heat losses of Housing Division homes by 85 per cent. Since insulation can be fitted at the owner's option, it is not considered as a difference in design. In evaluating design differences, it will be assumed that insulation to the same standard is fitted in the two Jennings houses. For optimum comparability, identical building materials will also be assumed.

Brick veneer construction is used in both public and private housing, and is the logical choice of a "standard" material. However, due to a slip at the computer terminal, the brick veneer house was given a 5 kW heater which was unable to maintain the required temperature on the day in question. As a result, the block veneer house will be used as a basis for comparison.

Table 4.5: Net heat flow by paths, on a winter's day, and relevant parameters of design: concrete block house, Hobart; heated area 49.25 m^2; $20^{o}C \pm 1^{o}C$, 7 a.m. - 11 p.m.; ceiling insulated.

General Parameter	A			$\sqrt{A}$				
Path of heat flow	Heat loss through infiltration	Heat loss by conduction		Heat loss by conduction			Solar heat gain through windows	Internal heat gains
		Ceiling	Floor	Internal walls	External walls	windows		
Specific parameter	A x $\bar{c}$	$A(m^2)$	$A(m^2)$	$L_i(m)$	$A_e(m^2)$	$A_w(m^2)$	$A_w(m^2)$	
Value of specific parameter	80.3	49.25	49.25	16.2	41.41	11.63	11.63	
Heat loss (kWh)	11.7	3.9	16.9	11.5	18.3	13.6	-4.4	-14.4
Net heat loss (kWh)	32.5			39.0				-14.4

Explanation of symbols: A = floor area of heated space.

$\bar{c}$ = mean of room air change rates, weighted according to room area (heated zone only).

L_i = Length of internal wall dividing heated and unheated zones, measured at wall base.

A_w = window area (heated zone).

A_e = Area of external wall enclosing heated space, excluding windows.

Since identical building materials are assumed, the various heat losses of the type 314 house (Table 4.6) can be scaled up or down for the three other houses, according to the measured parameters of the respective designs. These calculated heating loads are compared, in Figure 4.12, with the "assumed" and "ideal" heating loads, on an expanded version of Figure 4.11. The heating load of the second Housing Division house is very close to the predicted value, but the McArthur design has a heating load 3 per cent below the "assumed" load and 6 per cent below the "ideal" load, while that of the Mayfair design is 6 per cent below the "assumed" load, and 8 per cent below the "ideal" load.

These differences are primarily due to the inclusion of hallways in the heating areas of the Housing Division houses. Hallways have higher infiltration rates, and a higher ratio of wall area to floor area. Hence, they have greater heat losses per unit area.

The results show that heating loads can be assumed proportional to floor area, with likely errors up to 10 per cent.

4.5 SUMMARY AND CONCLUSIONS

The heating requirements of houses depend very much on the thermal properties of the house being heated, and can be accurately calculated by computer.

A range of housing materials and insulation levels has been selected, to represent the range of houses likely to be encountered in Tasmania. The results of heating load calculations will be presented in Chapter Six. To apply these results to other houses, both peak and annual heating loads can be assumed proportional to the floor area of the heated space. House size and design will in general have little effect on the economics of heat pumps, since changes in required heater capacity are compensated for by changes in total heating loads.

Table 4.6: Comparison of heating requirements of four houses: summary of calculations (winter's day, Hobart).

House design:	314	305	McArthur	Mayfair
Area of heated space (m^2):	49.25	41.42	33	40.54
Infiltration heat loss:				
$\bar{c}$ (hr^{-1})	1.63	1.43	2	1.23
$A \times \bar{c}$ ($m^2.hr^{-1}$)	80.28	59.23	66	49.86
heat loss (kWh)	11.7	8.6	9.6	7.3
Conducted heat loss through ceiling:				
A (m^2)	49.25	41.42	33	40.54
heat loss (kWh)	3.9	3.3	2.6	3.2
Conducted heat loss through floor:				
A (m^2)	49.25	41.42	33	40.54
heat loss (kWh)	16.9	14.2	11.3	13.9
Conducted heat loss through internal walls:				
L_i (m)	16.2	15.7	5.4	5.4
heat loss (kWh)	11.5	11.2	3.8	3.8
Conducted heat loss through external walls:				
A_e (m^2)	41.41	44.62	36.56	39.08
heat loss (kWh)	18.3	19.7	16.2	17.3
Net heat loss through windows:				
A_w (m^2)	11.63	7.22	8.56	10.48
heat loss (kWh)	9.2	5.7	6.8	8.3
Incidental heat gains: (kW)	14.4			
Heating deficit to be supplied by heater: (kWh)	57.1	48.3	35.9	39.4
Heating deficit excluding heat passed through interior walls to sleeping zone: (kWh)	45.6	37.1	32.1	35.6

For explanation of symbols, refer to Table 4.5.

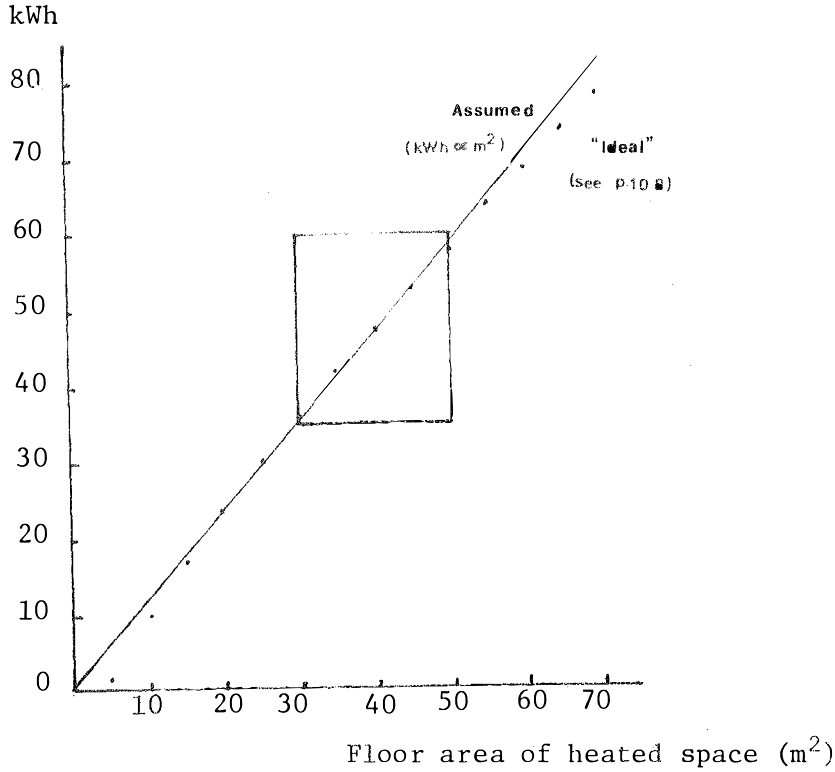


Figure 4.11: Comparison of "ideal" and assumed relationship between heating load and heated area.

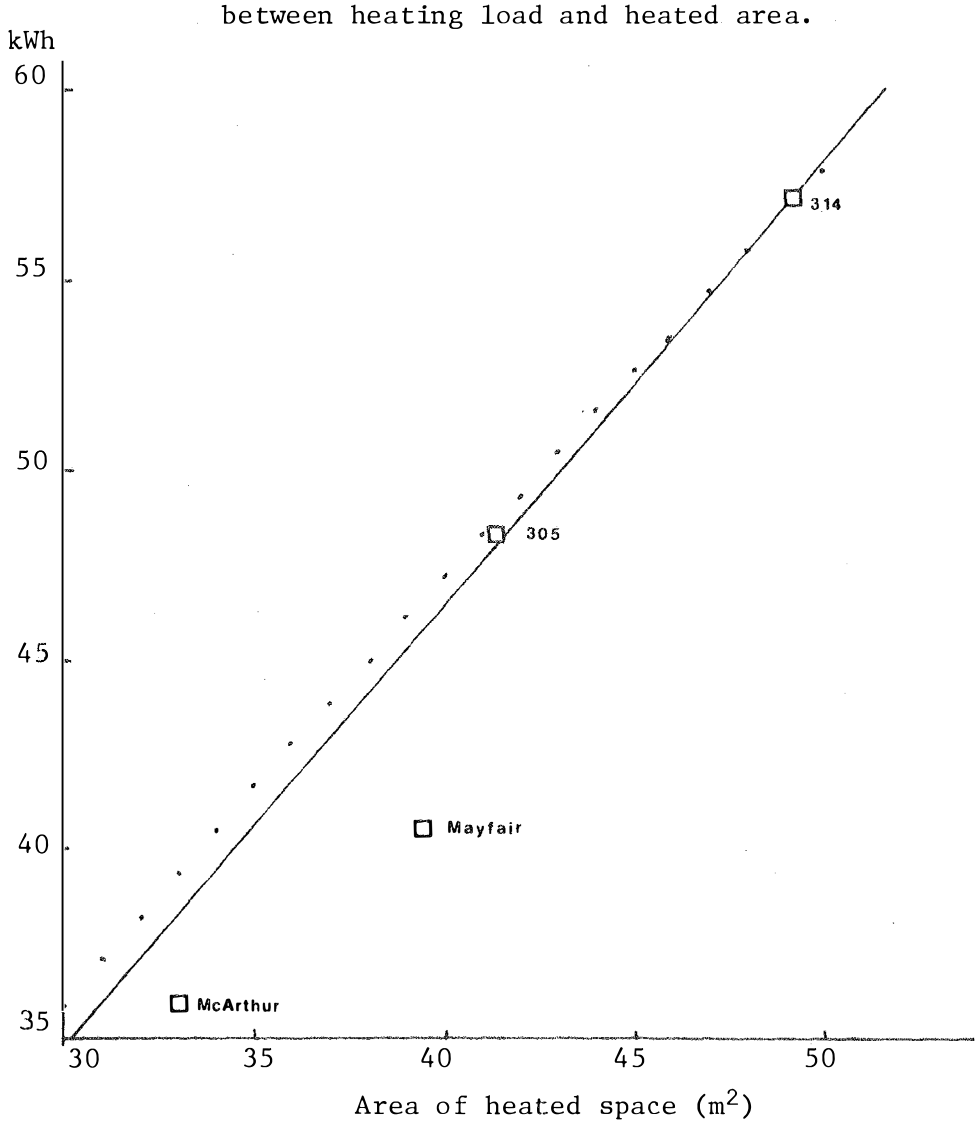


Figure 4.12: Heating requirements of four house designs

CHAPTER FIVE: HEATING ENERGY REQUIREMENTS

5.1 INTRODUCTION

Chapter Three has shown how comfort standards are related to Tasmania'a climate, and has derived heating standards for both the living and sleeping areas of Tasmanian homes. Chapter Four described a method by which the energy cost of maintaining these standards can be calculated. This method has been used to calculate heating loads for maintaining thermal comfort in the living areas of a number of houses.

Section 5.2 presents the heat energy requirements of each of the case studies. It also illustrates the variations in heating loads caused by differences in house materials, insulation and daily heating duration. The method of calculating the electrical loads of heat pump installations is described in section 5.3, and the seasonal efficiencies of other installations are considered in section 5.4. Annual energy requirements and annual running costs of all heating systems are compared in section 5.5.

In Chapter Six, running costs will be considered in conjunction with initial and maintenance costs, to determine the economic feasibilities of the heat pump installations.

5.2 HEATING REQUIREMENTS: CASE STUDIES

The TEMPAL computer programme has been used to calculate achieved temperatures, heating loads and seasonal heating energy requirements for six case studies involving intermittent heating to the living areas of houses in Hobart. The specifications of the six cases are listed in Table 5.1. Heating Comparison Charts (Figures 5.1, 5.3, 5.4) compare annual load factors, and show the performance of the chosen heater capacities in cold conditions, when the outdoor temperature falls below Hobart's design temperature of $3^{o}C$.

Table 5.1: Case studies: intermittent heating ($20^{o}C \pm 1^{o}C$) to living area (Heater capacities for both house designs are based on the recommendations of Coldicutt *et al.* (1978), for house design 305; curtains are assumed open by day and drawn at night.)

Case	House Design	Area Heated m^2	MATERIALS Floor	Roof	Walls	Insulation[1]	Period of heating[2]	Heater capacity kW	Seasonal energy requirement GJ
A	305	41.4	suspended timber	corrugated iron	brick veneer	ceiling	day/evening	5	31.7
B	314	49.25	"	concrete tile	concrete block veneer	"	"	7	32.4
C	314	49.25	"	corrugated iron	weatherboard	"	"	7	31.1
D	305	41.4	"	"	brick veneer	"	evening	5	9.8
E	305	41.4	"	"	"	ceiling/walls	day/evening	5	23.0
F	314	49.25	"	"	weatherboard	none	"	7	47.4

[1] Insulation: "ceiling" refers to the use of 100 mm loose-fill fibreglass insulation plus RFL (∿ 75 mm fibreglass batts); "walls" refers to the use of double-sided RFL, disked between the studs, in external walls (∿ 50 mm fibreglass batts).

[2] Period of heating: "day/evening": 7 a.m. - 11 p.m.
"evening": 5 p.m. - 11 p.m.

Table 5.2: Sample of computer output
(Case A)

LOADS AND ASSOCIATED EXTERNAL AIR TEMPERATURES
HOURS OF OCCURRENCE (HEATED ZONE)

EXTERNAL AIR TEMPERATURES, DEGREES CELSIUS

LOAD KW	< -3	-3 < 0	0 < 3	3 < 6	6 < 9	9 < 12	12 < 15	15 < 18	18 < 21	> 21	TOTAL
0 < 1	0	0	0	3	63	225	512	499	227	136	1665
1 < 2	0	0	0	4	34	137	96	48	7	0	326
2 < 3	0	0	3	23	142	258	240	96	14	0	776
3 < 4	0	0	2	31	132	163	146	31	0	0	505
4 < 5	0	0	8	26	110	148	58	4	0	0	354
5	0	12	39	123	259	136	29	0	0	0	598
TOTAL	0	12	52	210	740	1067	1081	678	248	136	

Effects of Building Materials

The effects of different building materials (brick veneer; concrete block veneer; weatherboard) are shown by a comparison of case studies A, B and C. As might be expected, the weatherboard house (with its lower U-value) requires slightly less heating than the other houses. When the difference in heated areas is allowed for, the brick veneer house requires the greatest amount of heating. The differences involved are small, being of the order of 5 per cent.

Figure 5.1 shows the living zone temperatures obtained in each of these cases, over 2½ midwinter days. The effects of thermal storage are evident, though small. The lowest overnight temperature recorded in the concrete block veneer house is 0.7^{o}C higher than that in the weatherboard house. Greater improvements in comfort would require better insulation and more heavily constructed dwellings (e.g. concrete slab floors, solid brick/concrete internal partitions).

Effects of Heating Regime

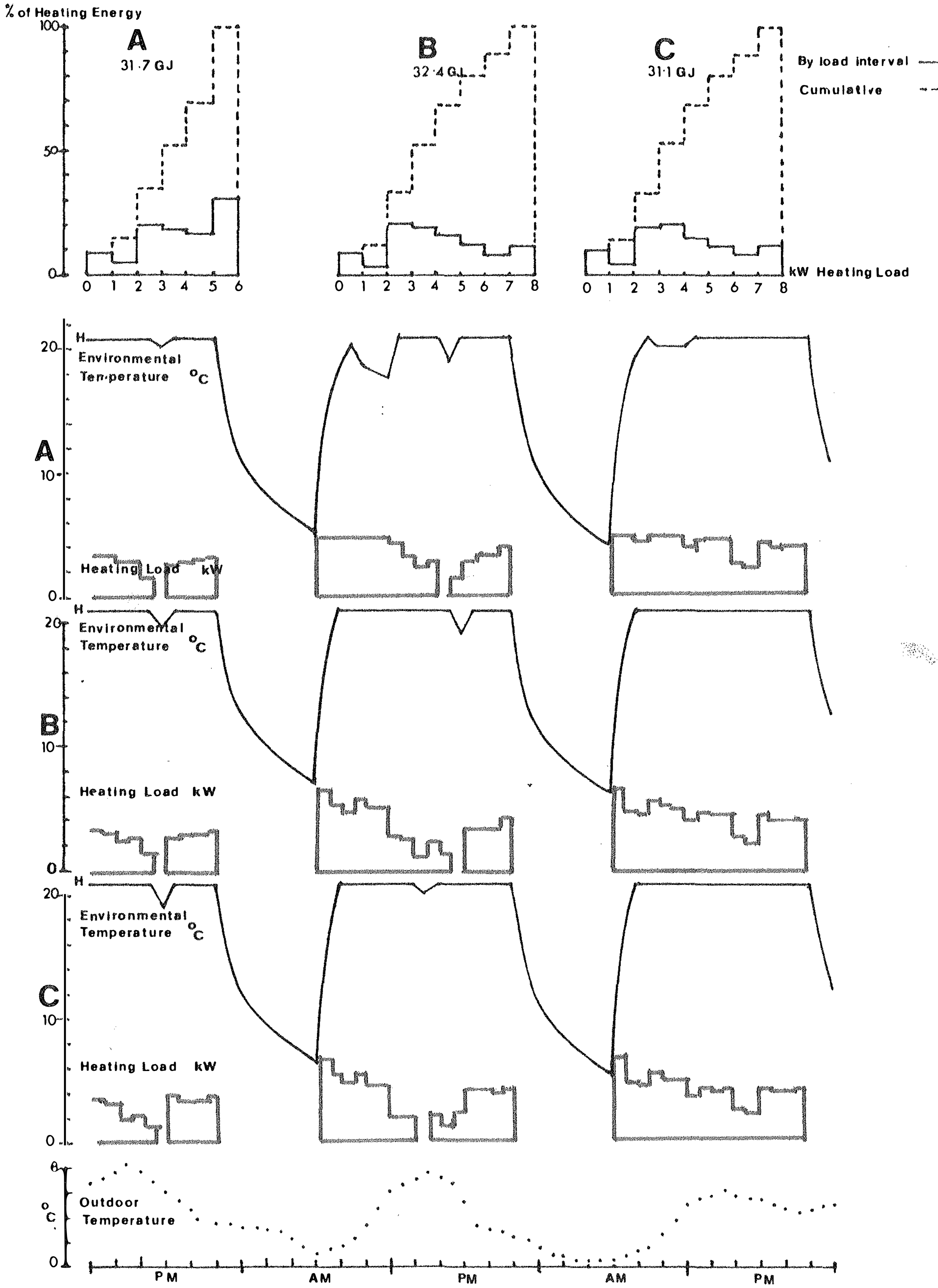
Temperature

Coldicutt *et al.* (1978) have calculated the annual energy requirements corresponding to inside temperatures of 18^{o}C ± 1^{o}C and 21^{o}C ± 1^{o}C, for the brick veneer house type 305B, with ceiling insulation. They are 20 GJ and 35 GJ respectively. This effect is depicted in Figure 5.2. Considerable energy savings can be made by reducing thermostat settings. This is most feasible during the day, when activity levels are usually higher. By reducing the daytime temperature to 15^{o}C, the energy saving could be as great as 7 GJ per year.

Duration of heating

The difference in energy totals (Figure 5.3) between day/evening heating and evening-only heating is quite marked. Evening-only heating uses less heating energy *per hour* than day-and-evening heating.

This appears to be due to the combined effects of solar radiation heat gain and thermal storage. Early morning temperatures and heat storage are at their daily minimum, and the heater must use a large amount of its capacity heating the living area from cold. In the late afternoon,



H: Heated (living) area

Figure 5.1: Heating Comparison Chart: effect of building materials.
Day-and-evening heating, insulated ceiling

- A: brick veneer/corrugated iron
- B: concrete block veneer/tile
- C: weatherboard/corrugated iron

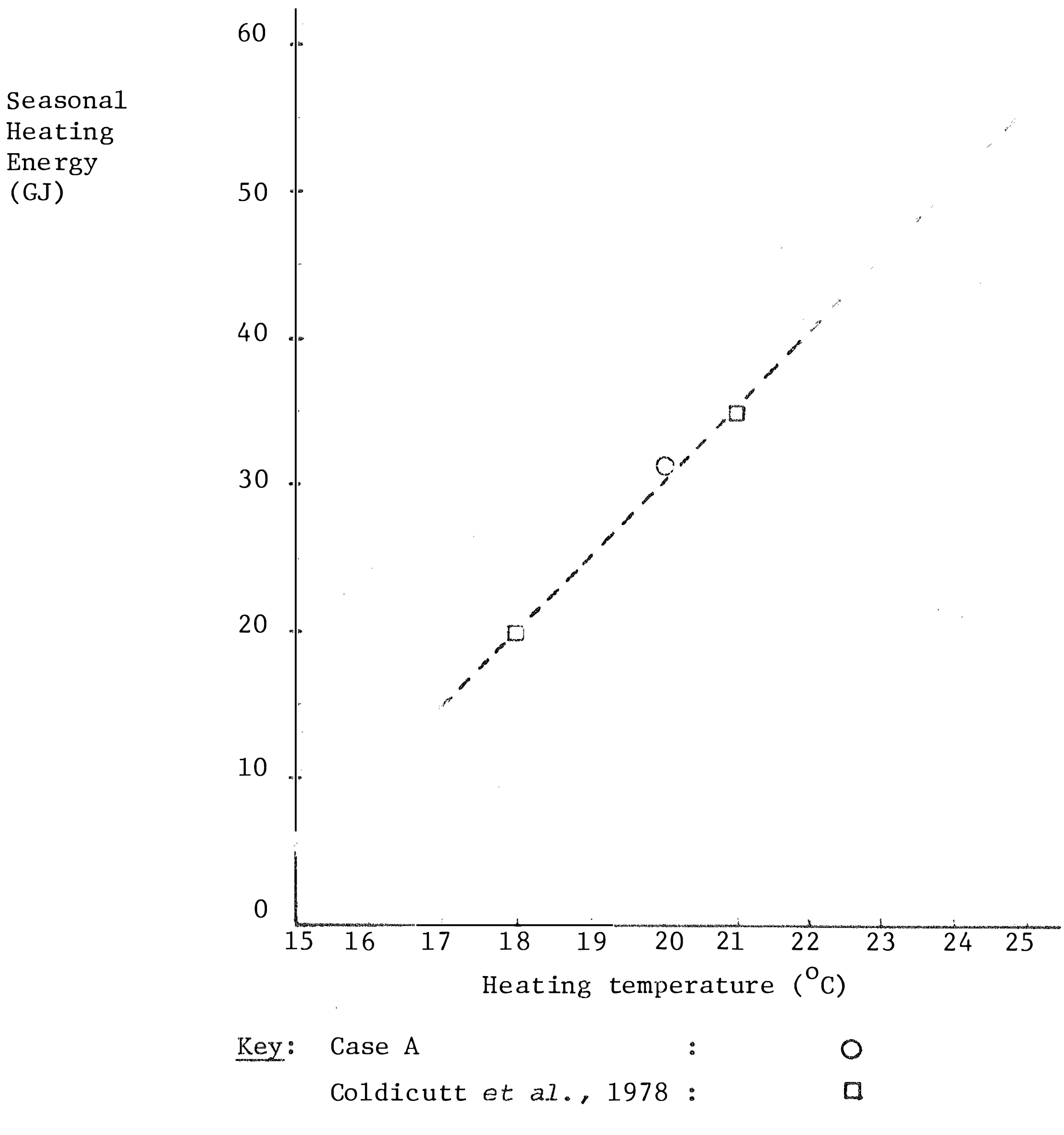
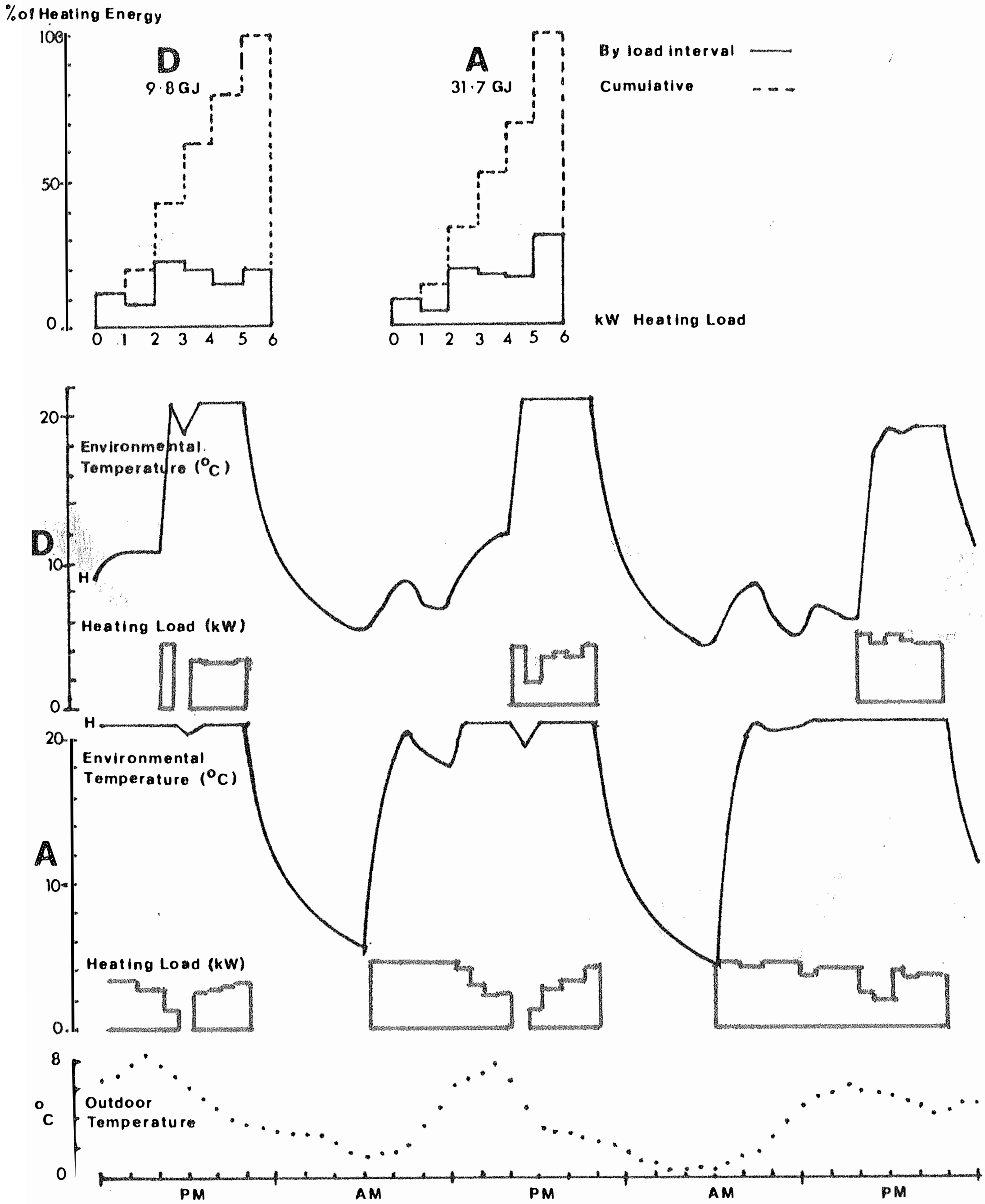


Figure 5.2: Effect of Heating Temperature on Seasonal Heating Energy Requirements; Day-and-evening heating, Brick Veneer House, insulated ceiling, Hobart



H: Heated (living) area

Figure 5.3: Heating Comparison Chart: effect of heating duration.

Brick veneer
Ceiling insulated

- D: evening only
- A: all day.

indoor and outdoor temperatures are higher, so that less energy is required in the initial heating period.

A second effect of this phenomenon is that evening heating can be done using a smaller heater. The 5 kW heater takes two hours to bring the living area to comfort levels in the morning, but only one hour at night.

It should be remembered, however, that the energy totals refer to the *total* duration of heating, so that any pre-heating time (using a time-switch) must be included in the estimation of energy requirements.

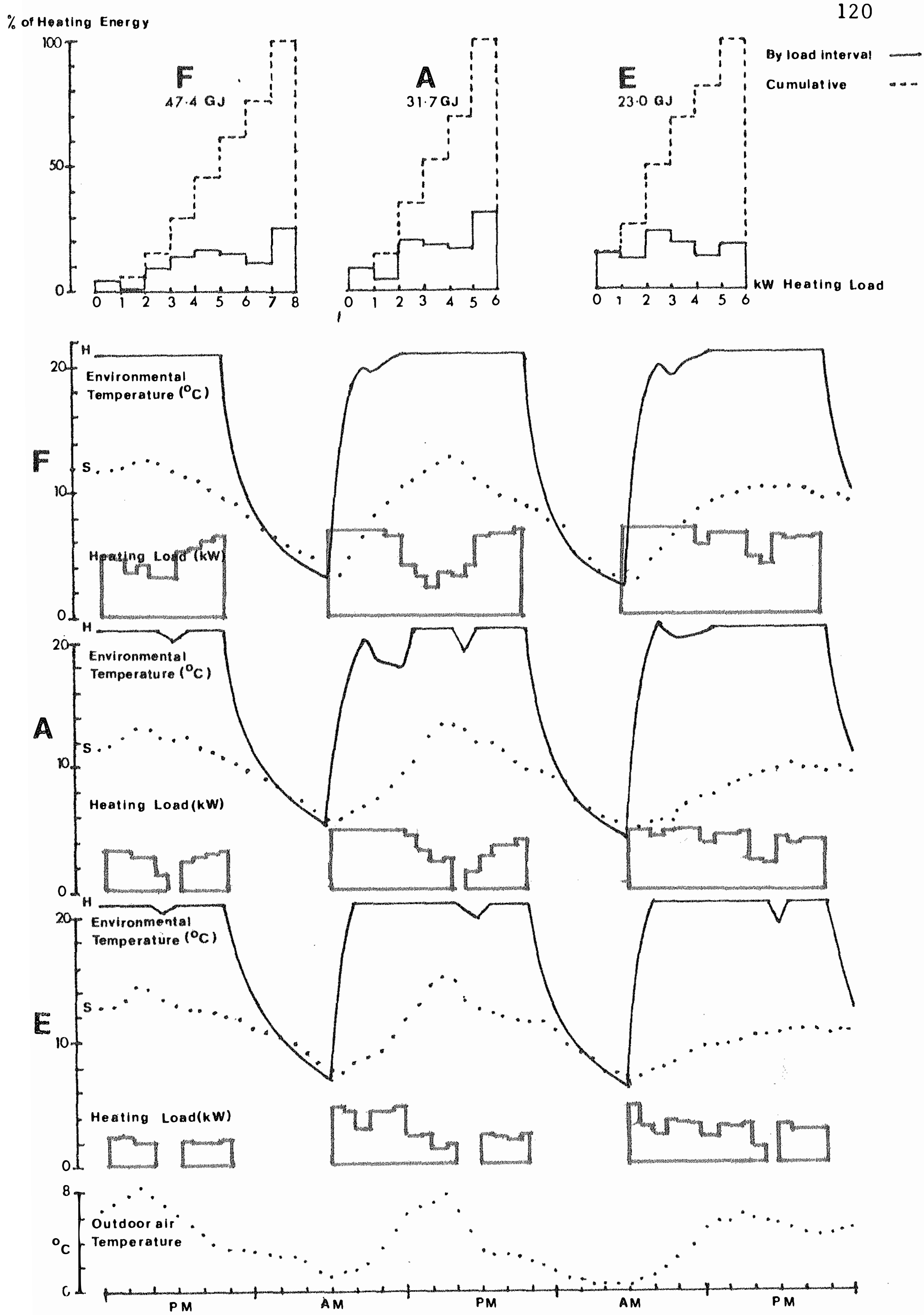
Effects of Insulation

As expected, insulation reduces both total energy requirements and plant capacity. The 7 kW heater in an uninsulated house (Figure 5.4) has a similar performance to the 5 kW heater in a house with ceiling insulation. With full insulation, the 5 kW heater warms up faster and spends less time overall (6.4 per cent) at full-load. During that time, it produces 18 per cent of the seasonal heating energy requirement, compared with 26 per cent (uninsulated, 7 kW) and 31 per cent (ceiling insulation, 5 kW).

Total energy requirements are reduced by an even larger factor than plant size. Ceiling insulation reduces the seasonal heating energy requirement from 47.4 GJ (case F) to 31.7 GJ (case A), while adding wall insulation results in a further reduction to 23 GJ (case E). This trend suggests that heat pumps may become uneconomic at high insulation levels, as the running cost savings are reduced faster than the savings in capital costs.

In a very real sense, heat pumps and insulation are in competition with each other. Even in terms of energy savings, the savings obtained by investing extra capital in a heat pump must be compared with the energy that could be saved by investing that money in additional insulation.

High levels of insulation improve the feasibility of a *changeover* heat pump system. In a well-insulated house, heat loss from the heated zone to the unheated zone has a greater heating effect on the (insulated) unheated zone. This is demonstrated in Figure 5.4. Sleeping zone temperatures over two winter days are consistently higher in the better



H: Heated (living) area
S: Sleeping area

Figure 5.4: Heating Comparison Chart: insulation: Day-and-evening heating

- F: No insulation (weatherboard)
- A: Ceiling insulated (brick veneer)
- E: Ceiling and walls insulated (brick veneer)

insulated dwellings. This means that when the heat pump switches from one zone to the other, less time will be needed to bring the latter zone to comfort levels.

5.3 POWER CONSUMPTION OF HEAT PUMP INSTALLATIONS

Heat pump capacities are normally chosen so that the heat pump provides a substantial fraction, but not all, of the peak heating load. The remainder of the peak load is taken up by resistance heating. Since the heat output of a heat pump increases with outdoor temperature, a heat pump large enough to supply the whole of the peak load would be grossly oversized for most of the heating season, when outdoor temperatures are higher than at peak times. A smaller heat pump, operating at a higher seasonal load factor, can still provide a large proportion of the heating load for a lower initial cost.

The computer was used to calculate the time for which each heating load was required at each outdoor temperature. An example of the output is shown (for case A) in Table 5.2. This information could then be used, in conjunction with manufacturers' data, to calculate the seasonal energy consumption and COP of any heat pump, for each of the six cases.

In most cases, energy requirements and COP's were determined for the full range of heat pumps (2.6 kW to 6.8 kW) with nominal heat outputs up to the maximum heating load. In the case of evening-only heating, the energy savings do not appear large enough to make even the smallest heat pump economical, and larger heat pumps were not considered.

To calculate the power requirements of each heat pump installation, heating loads were first grouped according to the associated outdoor temperatures. For each temperature range, the heat output, power consumption (including fan motors) and COP of the heat pump were determined from the manufacturer's data. Heating loads up to the output of the heat pump are taken to be supplied completely by the heat pump, and the energy consumed is the heat provided, divided by the COP. At higher loads, the heat pump works at full output, with the remainder of the heat provided by the supplementary resistance heating. The energy consumed by the heat pump is calculated from its rated power

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No allowance has been made for the reduction in COP which occurs during low load conditions due to frequent on/off cycling.

consumption at that temperature, multiplied by the time at the load-temperature combination. The power consumption of the resistance heater is equal to its heat output. The total heating requirement, the heat pump power consumption, and the resistance heating figures were each summed over the complete temperature range.

A worksheet for one of these calculations is shown as an example in Table 5.3. For all the air-conditioners, which were only assumed to have two sets of performance figures (above and below $6^{o}C$ see p. 36), the work-sheets were considerably less complex.

Information from these worksheets was used to derive the information presented in Table 5.4. It should be noted that the "average COP of heat pump" figures refer only to the heat output and energy consumption of the heat pump. In the "seasonal COP" figures, the heat output and energy consumption of the resistance heating has been included.

"Value of annual energy savings" figures, based on the current marginal household tariff, have been included, as a guide only.

5.4 ENERGY REQUIREMENTS OF OTHER HEATER TYPES

Energy requirements of gas, oil, wood and direct electric heaters can be calculated from heating energy requirements, corrected for efficiency.

Electric storage heaters lose a certain amount of heat at all times. Figures 5.1, 5.3 and 5.4 show that most heat is lost from a house within two hours. Thus, the heat lost from a storage heater within two hours of the beginning of the heating period contributes to the required heating. Heat losses at other times may be considered to be wasted.

The heat loss for a fully insulated (6 kW) heat bank is assumed to be 5 per cent of the rated capacity, or 300 watts. For evening-only heating, the total amount of heat wastage is 4.8 kWh per day, or 4.6 GJ over the heating season. For day-and-evening heating, it is 1.8 kWh per day, and 1.7 GJ over the heating season.

An electric midi-bank (rated capacity 3.5 kW) stores 44 kWh over a 12½ hour charging period, and releases its heat continuously at an average rate of 1.8 kW. Its daily output is 44 kWh, of which 33 kWh contributes to day-and-evening heating. Because of its continuous heat

Table 5.3: Worksheet – Heating case F; 5.3 kW heat pump.

Temperature range (outdoor)	Heating capacity of heat pump	Load range	Time at load range	Mean load factor	Total heating energy	HEAT PUMP Electrical input	Heat provided	COP	Net electricity consumption	Supplementary heating
°C	kW	kW	hours	%	kWh	kW	kWh		kWh	kWh
-3 ~ 0	3.85	7	12	100	84	2.25	46.2	1.7	27	37.8
0 < 3	4.3	4 < 5	1	98.95	4.5	2.4	4.3	1.8	2.4	0.2
		5 < 6	2	100	11	2.4	219.3		122.4	132.7
		6 < 7	4		26					
		7	45		315					
3 < 6	4.75	2 < 3	1	53	2.5		9.5	1.9	5.0	–
		3 < 4	2	74	7					
		4 < 5	8	94.08	36	2.5	35.8		18.8	0.2
		5 < 6	30	100	165	2.5	945.25		497.5	381.3
		6 < 7	43		279.5					
		7	126		882					
6 < 9	5.2	0 < 1	9	1	4.5		795	2.0	397.5	–
		1 < 2	5	29	7.5					
		2 < 3	37	48	92.5					
		3 < 4	52	67	182					
		4 < 5	113	87	508.5					
		5 < 6	187	99.62	1028.5	2.55	968.7		475.0	59.8
		6 < 7	118	100	767	2.55	1752.4		859.4	547.6
		7	219		1533					
9 < 12	5.5	0 < 1	111	9	55.5		2408	2.05	1174.6	–
		1 < 2	14	27	21					
		2 < 3	111	45	277.5					
		3 < 4	250	64	875					
		4 < 5	262	82	1179					
		5 < 6	147	97.73	808.5	2.6	790.1		373.5	18.4
		6 < 7	77	100	500.5	2.6	946		447.2	219.5
		7	95							
12 < 15	5.7	0 < 1	388	9	194		2172	2.05	1059.5	–
		1 < 2	26	26	39					
		2 < 3	240	44	600					
		3 < 4	227	61	794.5					
		4 < 5	121	79	544.5					
		6 < 7	16	100	104	2.7	233.7		110.7	45.3
		7	25		175					
15 < 18	5.9	0 < 1	523	8	261.5		690	2.1	328.6	–
		1 < 2	14	25	21					
		2 < 3	90	42	225					
		3 < 4	38	59	133					
		4 < 5	11	76	49.5					
		5 < 6	2	93.14	11	2.75	11.0		5.1	0.0
18 < 21	6.0	0 < 1	246	8	123	2.9	128	2.06	62.1	–
		2 < 3	2	42	5					
> 21	6.0	0 < 1	135	8	67.5	2.9	80	2.07	38.65	–
		2 < 3	1	[illegible]	2.5					
					13877		12442.55		6103.15	1444.5

Average COP of heat pump: 2.04

Percentage of total heating energy supplied by heat pump: 90%

Percentage of heating provided by heat pump below 3°C: 61%

Seasonal COP: 1.84

Annual energy savings: 46 per cent of 47.4 GJ = 21.6 GJ

Value of annual energy savings: $181

Table 5.4: Energy requirements of heat pumps: case studies.

Case	Heat pump	average COP of heat pump	% of total heat supplied	% of heating below 3°C [2]	Seasonal COP	Total electricity consumption	Annual energy savings	Value of energy saving[1]
A 5 kW 31.7 GJ	2.6 kW	2.28	70		1.65	19.2	12.5 GJ	$105
	3.8 kW	2.11	85		1.81	17.5	14.2 GJ	$119
	5.3 kW	2.06	99	86	2.04	15.5	16.2 GJ	$135
B 7 kW 32.4 GJ	2.6 kW	2.33	67		1.61	20.1	12.3 GJ	$103
	3.8 kW	2.11	80		1.73	18.7	13.7 GJ	$115
	5.3 kW	2.02	94	68	1.91	17.0	15.4 GJ	$129
	6.8 kW	2.51	98	81	2.45	13.2	19.2 GJ	$160
	6.4 kW	1.92	95		1.84	17.6	14.8 GJ	$124
C 7 kW 31.1 GJ	2.6 kW	2.32	67		1.61	19.3	11.8 GJ	$ 99
	3.8 kW	2.10	80		1.73	20.0	13.1 GJ	$109
	5.3 kW	2.02	94	68	1.90	16.3	14.8 GJ	$124
	6.8 kW	2.51	98	81	2.44	12.7	18.4 GJ	$154
	6.4 kW	1.91	95		1.83	17.0	14.1 GJ	$118
D 5 kW 9.8 GJ	2.6 kW	2.31	75		1.73	5.65	4.2 GJ	$ 34
E 5 kW 23.0 GJ	2.6 kW	2.31	77		1.78	12.9	10.0 GJ	$ 84
	3.8 kW	2.09	89		1.86	12.4	10.6 GJ	$ 89
F 7 kW 47.4 GJ	2.6 kW	2.35	55		1.46	32.5	15.0 GJ	$126
	3.8 kW	2.13	71	> 29	1.61	29.4	18.04 GJ	$151
	5.3 kW	2.04	90	61	1.84	25.8	21.6 GJ	$181
	6.8 kW	2.55	97	75	2.45	19.3	28.03 GJ	$235
	6.4 kW	1.95	93		1.83	25.9	21.5 GJ	$180

[1] Value of energy saving is calculated at the household tariff electricity rate of $8.37 per GJ.

[2] Proportion of heating supplied by heat pump at outdoor temperatures below 3°C (per cent).

loss, this type of heater is unsuitable for evening-only heating. Also, in the case of the uninsulated house, the greater heating load justifies the use of the more sophisticated heat bank in preference.

The total electrical energy consumption of the midi-bank system is found by adding the supplementary heating requirement to the electrical consumption of the heat bank (44 kWh per day, or 41.9 GJ for the heating season). The electrical energy consumption of the heat bank system can be estimated by adding the heat wastage (4.6 GJ or 1.7 GJ) to the seasonal heating energy requirement.

The electrical energy requirements of electric storage heating systems are listed in Table 5.5.

Table 5.5: Electrical energy requirements of storage heating systems.

Heating Case:	A	B	C	D	E	F
Heating energy requirement (GJ):	31.7	32.4	31.1	9.8	23.0	47.4
HEAT BANK (6 kW)						
Useful heat supplied (GJ)	31.0	28.5	29.8	9.8	22.8	41.3
Heat wasted (GJ)	1.7	1.7	1.7	4.6	1.7	1.7
Supplementary heating (GJ)	0.7	-	1.3	-	0.2	6.1
TOTAL (GJ)	33.4	34.1	32.8	14.4	24.7	49.1
MIDI BANK[1] (3.5 kW)						
Total off-peak (GJ)	41.9	41.9	41.9	-	41.9	41.9
Supplementary heating (GJ)	8.2	9.7	8.7	-	4.1	20.0
TOTAL (GJ)	50.1	51.6	50.6	-	45.0	61.9

[1] These figures also apply to a "combi-bank" which combines a passive 3.5 kW off-peak heater with direct heating.

For comparison of the primary energy requirements of different heating systems, the energy use of each system has been calculated for case C. The results are shown in Figure 5.5.

Both forms of heating will need supplementary direct heating when the

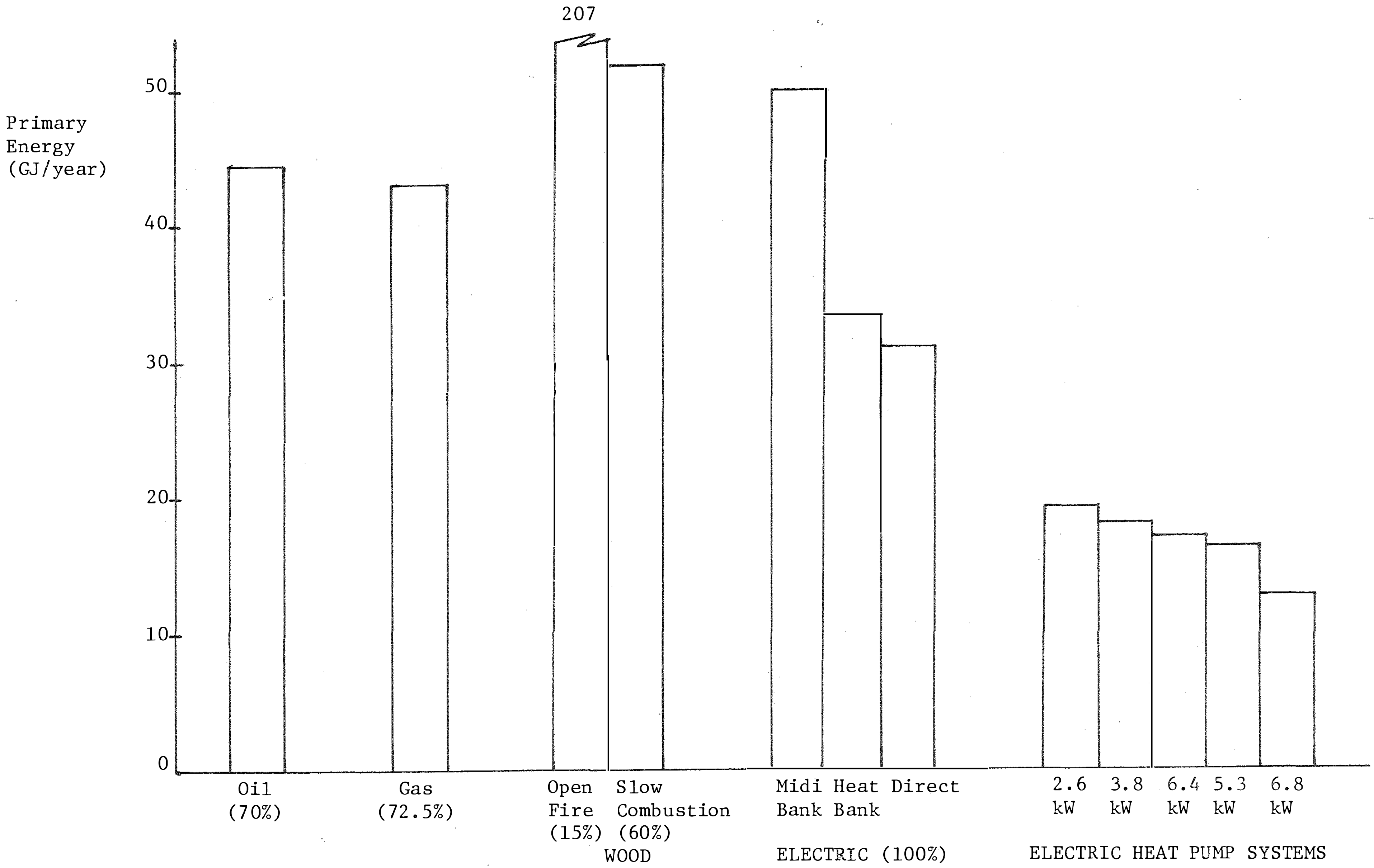

Figure 5.5.: Primary Energy used by heating systems providing day-and-evening heating (20°C) to the living zone of an insulated (ceiling only) weatherboard house in Hobart. Assumed efficiencies given in brackets.

daily heating demand exceeds the effective storage capacity. For evening-only heating, the 6 kW heat bank has a storage capacity of 48 kWh on an 8-hour charge. For day-and-evening heating, with a 2½-hour recharging period in the afternoon, the effective daily output increases to 64 kWh. The effective daily output of the midi-bank, for day-and-evening heating, is 33 kWh.

Supplementary heating requirements have been calculated from the daily heating loads calculated by computer.

The energy requirements can also be used to calculate seasonal running costs. Since electricity is charged at the meter, it is taken to be delivered as heat (for costing purposes) at 100 per cent efficiency. Requirements of other fuels must be corrected for efficiency of heating.

Annual running costs (at June 1979 prices) are listed in Table 5.6. These costs will be used in the economic evaluation carried out in the next chapter.

5.5 SUMMARY

Computations of heating loads have highlighted the reductions in heating energy requirements that can be achieved through the use of insulation, and the effect of daily heating duration on energy requirements. Seasonal heat pump COP's, calculated from the computer results, range from 1.46 to 2.45, with corresponding energy savings when compared to direct electric resistance heating.

Primary energy requirements of heating systems vary widely. The major points of interest here are the high primary energy cost of heating in an open fireplace, the considerable energy wastage of poorly insulated or uninsulated electric storage heaters, and the low primary energy consumption of heat pumps.

Table 5.6: Annual running costs of heating systems ($/year, June 1979 prices)

Primary heater	Case A	Case B	Case C	Case D	Case E	Case F
Oil heater[1]	246	252	242	76	179	369
Gas heater[2]	332	339	326	103	241	496
Open fireplace[3]	316	323	310	98	229	472
Slow combustion[4]	96	98	94	30	70	144
Electric						
- direct[5]	265	271	260	82	192	397
- midi-bank (3.5 kW)[6]	234	246	238	~165	200	333
- heat bank (6 kW)[6]	138	144	138	61	100	221
Heat pump - 2.6 kW[5]	187	168	161	47	108	272
3.8 kW[5]	146	157	151	-	104	246
6.4 kW[5]	-	147	142	-	-	217
5.3 kW[5]	130	142	136	-	-	216
6.8 kW[5]	-	110	106	-	-	162

1 70 per cent efficiency, 17.58 cents per litre of heating oil.

2 72.5 per cent efficiency, $17 per 45 kg cylinder of L.P. gas.

3 Assuming 15 per cent efficiency, $23 per tonne of firewood (18").

4 60 per cent efficiency, $28 per tonne of firewood (9").

5 Marginal household tariff 3.01 ¢/kWh (including tax).

6 Supplementary heating: as for note 5.
Off-peak: first 2000 kWh/quarter at 1.52 ¢/kWh (including tax).
marginal rate 1.32 ¢/kWh (including tax).

CHAPTER SIX: ECONOMIC EVALUATION AND GENERAL CONCLUSIONS

6.1 INTRODUCTION

This chapter combines the results of heating energy calculations (Chapter Five) with information about initial and maintenance costs (Chapter Two) in order to determine the net costs of both heat pumps and conventional space heaters (section 6.2).

By comparing net costs for each heating case, the applications for which heat pumps are economic can be determined (section 6.3). The climatic adjustment factor derived in Chapter Three can be used to determine the economic viability of heat pumps for locations other than Hobart. From the list of economic applications, the types of heat pump most suited to the household heating market can be found, and the likely place of heat pumps in the market predicted (section 6.4).

The economic analysis permits the cost-effectiveness of heat pumps, as a means of conserving energy, to be compared with that of insulation (section 6.5). Sources of uncertainty in the economic comparison are considered in section 6.6, whilst sections 6.7 and 6.8 discuss the implications of this study with respect to the Hydro-Electric Commission and the Housing Division. General conclusions are presented in section 6.9.

6.2 CALCULATION OF NET COSTS

The method used for calculation of net costs is the *net present value* (NPV) method. Present costs are counted at full value, and future costs are discounted at a compound interest rate. In the case of heating, all major items are costs, and to save confusion the calculated net cost of a heating system will be referred to as its *net present cost* (NPC). The most economic heating system will be the one with the lowest NPC.

In order to carry out this type of analysis, it is necessary to specify what costs occur, and when they occur. It is also necessary to determine the appropriate discount rate to be applied to future costs. The means by which the relevant parameters have been determined will now be described.

Discount Rate

In this analysis, two values of discount rate - representing the expected upper and lower limits for investment in a heat pump - will be used. They are:

(i) the interest rate on long-term Government bonds (9.25 per cent per annum); and

(ii) the bank interest rate on a secured ten-year loan (14.4 per cent per annum).

These values have been chosen for practical reasons. The interest rate on long-term Government bonds represents the marginal rate at which people will invest their money. It is set sufficiently high to attract investment, but as low as possible in order to minimise the interest cost to the Government.

This rate applies to a person with sufficient cash on hand to have a free choice of heating systems. This person has two types of options (assuming that there is a range of heater types that can fully meet his heating requirements):

(a) he can buy an expensive heater that is cheap to run (e.g. a heat pump);

(b) he can buy a cheaper heater that is more expensive to run (e.g. direct electric heating).

[Note that the ideal situation - a cheaper heater with cheaper running costs - will prove economically superior whatever discount rate is used.]

He can make a rational economic decision by comparing the difference in running costs with the interest he could obtain by investing the surplus capital of case (b). In this decision-making process, future costs are implicitly discounted at the investment interest rate of 9.25 per cent.

If, on the other hand, he must borrow the money to buy the heater, the economic decision rests on a comparison of the combined loan repayments and running costs. In this case, future costs are discounted at the bank interest (i.e. compound interest) rate of 14.4 per cent per annum.

The appropriate discount rates for intermediate situations (in which a part of the initial cost is borrowed) will lie between these two limits.

It should be noted that these discount rates allow for the effects of inflation, and are to be applied to inflated running costs. Hence they are higher than those used to deal with running costs estimated at present dollar values.

Depreciation of Initial Costs

Because heaters have varying lifetimes, generally between ten and twenty years, their depreciation times must be taken into account.

Akalin (1978) lists depreciation periods and equipment life statistics for heating and cooling equipment. The twelve residential heat pumps in his survey had a mean lifetime of eleven years (median and modal lifetimes both ten years). Twenty-eight gas or electric unit heaters averaged fourteen years' lifetime (median thirteen, mode ten), whilst items of air handling equipment (ducts, fans, etc.) had lifetimes around twenty years. Minimum depreciation times given were twenty years (air terminals), fifteen years (gas and electric heaters) and ten years (oil burner equipment; air-conditioning systems under 16 kW output).

These data have been used as a guide in estimating the depreciation times used for domestic heating equipment in Tasmania (Table 6.1).

Costs of heating are compared over a ten-year period. Thus, the initial costs of oil heaters and heat pump units will be fully depreciated.

For simplicity, the initial costs of longer-lived heaters are depreciated in inverse proportion to their lifetimes. For example, two-thirds of the initial costs of electric heaters are assumed to be depreciated over the ten-year period.

Normally, future use is discounted, so that the depreciation cost would be greater than that assumed here. However, since running costs in most cases are much greater than initial costs, the difference made by discounting will in general be small. Only the off-peak heat bank and open fireplace will be favoured to any significant extent by this practice. [If future use is to be discounted at 5 per cent per annum, the depreciated proportion of the initial cost of electric heating rises from 67 per cent to 75 per cent; for the open fireplace it rises from 25 per cent to 46 per cent (i.e. from $200 to $368).]

Table 6.1: Depreciation times allowed for heating equipment.

Electric heaters	15 years
Gas heaters	15 years
Slow combustion heaters	15 years
Open fireplace	40 years (lifetime of house)
Oil heater	10 years
Heat pumps:	
heat pump unit	10 years
ducting	20 years
thermostat	20 years
supplementary heating	15 years

Recurrent Costs

Estimated price inflation rates for fuels have been derived in Appendix C. Maintenance costs are assumed to increase according to the Consumer Price Index, which rose by a factor of 2.05 between 1969 and 1977 (Hartley, Jones and Badcock, 1978). Adding inflation rates of 9.6 per cent and 8.8 per cent for 1978 and 1979 gives a rise of 2.45 over ten years. The ten-year mean annual CPI inflation rate of 9.4 per cent will be assumed for future maintenance costs.

In calculating net present costs, these inflated costs are discounted according to the discount rate being applied.

Current prices of installation, maintenance and fuels, for each heating case, are shown in Table 6.2.

As an example of the NPC calculation, consider the use of direct electric heating in the brick veneer house, case A (ceiling insulated, heating $20^{o}C \pm 1^{o}C$ to living zone, from 7 a.m. to 11 p.m.), using the 14.4 per cent discount rate:

(i) Depreciation is assumed to be two-thirds of the estimated initial cost of \$350, or \$233.

(ii) Maintenance, at \$5 per year, is assumed to increase at 9.4 per cent per annum, giving annual maintenance costs of \$5.47, \$5.98, \$6.55, \$7.16 .... in successive years. The present costs of maintenance, discounted at 14.4 per cent per annum, are \$4.78, \$4.57, \$4.37 ...., summing to a net present cost, over ten years, of \$39.43.

(iii) Electricity costs, inflating at 11.56 per cent per annum, will be \$296, \$330, \$368 .... in successive years, with discounted present costs of \$258, \$252, \$246 .... summing over ten years to a net present cost of \$2314.

The final NPC of direct electric heating, at the 14.4 per cent discount rate, is found, by adding net present depreciation, maintenance and electricity costs, to be \$2586.

Values of present costs (Table 6.2) have been used to calculate Net Present Costs of the range of heating systems for each of the six heating cases described in Chapter Five (Table 6.3). For ease of comparison, these costs are represented graphically in Figures 6.1 to 6.6.

6.3 ECONOMIC COMPARISON

The results of the NPC calculations (Figures 6.1 to 6.6) show that in general heat pumps are competitive with most conventional forms of heating. In each case involving day-and-evening heating, the most economic heat pump system had a lower NPC than oil, gas or wood heating using an open fire. In all cases, both the electric heat bank system and the slow combustion heater proved cheaper than the heat pump systems. For evening-only heating, the total net costs of most systems fell below the combined net initial and maintenance costs of the smallest heat pump under consideration, making this use uneconomic for heat pumps in general.

The relative economics of the 3.5 kW midibank appear to depend on the design heating load, which in turn depends on insulation (cf. cases C, E, F) and house size. This implies that the economics of this type of storage heater depend strongly on choosing the optimum rated capacity of off-peak heating. The economics of the midibank may be improved in insulated houses by the choice of a smaller model. For this reason, it is not certain that heat pumps will be more economic, in any case of day-and-evening heating, than a properly chosen midibank.

Table 6.2: Summary of costs (1979 prices).

Type of heater:	OIL	GAS	WOOD		ELECTRIC			HEAT PUMP				
			slow combustion	open fire	midi bank	heat bank	direct	through-the-wall			unitary	
Nominal capacity (kW)	10.8	5/7.4	13	?	3.5	6	5/7	2.6	3.8	6.4	5.3	6.8
Depreciation period (years)	10	15	15	40	15	15	15	10[1]	10[1]	10[1]	10[1]	10[1]
Total installed cost ($)												
- cases A,D,E[5]	500	450	400	800	500[2]	800[3]	350	950[2]	1100[2]	-	1600[4]	-
- cases B,C,F[5]	500	490	400	800	600[2]	900[2]	490	1050[2]	1200[2]	1450[2]	1650[4]	2000[4]
10-year depreciation cost ($)												
- cases A,D,E[5]	500	300	267	200	333	533[3]	233	850	1030	-	1500	-
- cases B,C,F[5]	500	327	267	200	400	600	327	920	1070	1350	1530	1750
Annual maintenance cost ($)	20	15	10	8	5	5	5	30	45	60	60	70
Annual fuel cost ($)												
- case A[5]	246	332	96	316	234	138	265	187	146	-	130	-
- case B[5]	252	339	98	323	246	144	271	168	157	147	142	110
- case C[5]	242	326	94	310	238	138	260	161	151	142	136	106
- case D[5]	76	103	30	98	-	61	82	47	-	-	-	-
- case E[5]	179	241	70	229	200	100	192	108	104	-	-	-
- case F[5]	369	496	144	472	333	221	397	272	246	217	216	162

1 Depreciation period 15 years for auxiliary heating; 20 years for thermostat and (where applicable) ducting.

2 Cost includes auxiliary direct electric heating, estimated at $50 per kilowatt.

3 Installed cost $750 (depreciation cost $500) for case D, which requires no auxiliary heating.

4 Price estimate for system with minimal ducting, or split system console unit. Normal ducting will add up to $800 to the installed cost, or up to $400 to the depreciation cost.

5 A,B,C,D,E,F refer to the case studies listed in Table 5.1, and depicted in Figures 6.1 to 6.6.

Table 6.3: Net present heating cost over ten years[1 2] at 9.25 per cent discount rate (costs at 14.4 per cent discount rate shown in brackets).

	Depreciation $	Maintenance $	Fuel $	Total $
CASE A:				
Oil - (24.4% p.a.)	500	202 (158)	5382 (4014)	6084 (4672)
- (12.9% p.a.)	"	"	2961 (2290)	3663 (2948)
Gas	300	151 (118)	3295 (2581)	3746 (2999)
Wood				
- slow combustion	267	101 (79)	984 (770)	1352 (1116)
- open fire	200	81 (63)	3241 (2533)	3522 (2796)
Electric				
- midibank	333	50 (39)	2633 (2045)	3016 (2417)
- heat bank	533	50 (39)	1553 (1206)	2136 (1778)
- direct	233	50 (39)	2979 (2314)	3262 (2586)
Heat Pump - 2.6 kW	850	302 (237)	2102 (1633)	3254 (2720)
- 3.8 kW	1030	453 (355)	1641 (1275)	3124 (2659)
- 5.3 kW	1500	605 (473)	1461 (1135)	3566 (3108)
CASE B:				
Oil - (24.4% p.a.)	500	202 (158)	5513 (4112)	6215 (4770)
- (12.9% p.a.)	"	"	3033 (2345)	3735 (3003)
Gas	327	151 (118)	3365 (2636)	3843 (3081)
Wood				
- slow combustion	267	101 (79)	1005 (786)	1373 (1132)
- open fire	200	81 (63)	3312 (2586)	3593 (2849)
Electric				
- midibank	400	50 (39)	2768 (2150)	3218 (2589)
- heat bank	600	"	1620 (1259)	2270 (1898)
- direct	327	"	3046 (2366)	3423 (2732)
Heat Pump - 2.6 kW	920	302 (237)	1888 (1467)	3110 (2624)
- 3.8 kW	1070	453 (355)	1765 (1371)	3288 (2796)
- 6.4 kW	1350	605 (473)	1652 (1283)	3607 (3106)
- 5.3 kW	1530	605 (473)	1596 (1240)	3731 (3243)
- 6.8 kW	1750	705 (552)	1236 (960)	3691 (3262)

Table 6.3: (Continued)

	Depreciation $	Maintenance $	Fuel $	Total $
CASE C:				
Oil - (24.4% p.a.)	500	202 (158)	5294 (3949)	5996 (4607)
- (12.9% p.a.)	"	"	2912 (2252)	3614 (2910)
Gas	327	151 (118)	3236 (2535)	3714 (2980)
Wood				
- slow combustion	267	101 (79)	964 (754)	1332 (1100)
- open fire	200	81 (63)	3179 (2485)	3460 (2748)
Electric				
- midibank	400	50 (39)	2678 (2080)	3128 (2519)
- heat bank	600	"	1553 (1206)	2203 (1845)
- direct	327	"	2922 (2270)	3299 (2636)
Heat Pump - 2.6 kW	920	302 (237)	1810 (1406)	3032 (2563)
- 3.8 kW	1070	453 (355)	1697 (1318)	3220 (2743)
- 6.4 kW	1350	605 (473)	1596 (1240)	3551 (3063)
- 5.3 kW	1530	605 (473)	1529 (1187)	3664 (3190)
- 6.8 kW	1750	705 (552)	1191 (925)	3646 (3227)
CASE D:				
Oil - (24.4% p.a.)	500	202 (158)	1663 (1240)	2365 (1898)
- (12.9% p.a.)	"	"	915 (707)	1617 (1365)
Gas	300	151 (118)	1022 (801)	1473 (1219)
Wood				
- slow combustion	267	101 (79)	308 (246)	676 (586)
- open fire	200	81 (63)	1005 (786)	1286 (1049)
Electric				
- heat bank	500	50 (39)	686 (533)	1236 (1072)
- direct	233	"	922 (716)	1205 (988)
Heat Pump - 2.6 kW	850	302 (237)	528 (410)	1680 (1497)
CASE E:				
Oil - (24.4% p.a.)	500	202 (158)	3916 (2921)	4618 (3579)
- (12.9% p.a.)	"	"	2154 (1666)	2856 (2324)

Table 6.3: (Continued)

(CASE E: contd.)	Depreciation $	Maintenance $	Fuel $	Total $
Gas	300	151 (118)	2392 (1874)	2843 (2292)
Wood				
- slow combustion	267	101 (79)	718 (561)	1086 (907)
- open fire	200	81 (63)	2348 (1836)	2629 (2099)
Electric				
- midibank	333	50 (39)	2250 (1748)	2633 (2120)
- heat bank	533	"	1125 (874)	1708 (1446)
- direct	233	"	2158 (1676)	2441 (1948)
Heat Pump - 2.6 kW	850	302 (237)	1214 (943)	2366 (2030)
- 3.8 kW	1030	453 (355)	1169 (908)	2652 (2293)
<u>CASE F</u>:				
Oil - (24.4% p.a.)	500	202 (158)	8073 (6021)	8775 (6679)
- (12.9% p.a.)	"	"	4441 (3434)	5143 (4092)
Gas	327	151 (118)	4923 (3856)	5401 (4301)
Wood				
- slow combustion	267	101 (79)	1477 (1154)	1845 (1500)
- open fire	200	81 (63)	4840 (3784)	5121 (4047)
Electric				
- midibank	400	50 (39)	3747 (2910)	4197 (3349)
- heat bank	600	"	2486 (1932)	3136 (2571)
- direct	327	"	4462 (3466)	4839 (3832)
Heat Pump - 2.6 kW	920	302 (237)	3057 (2375)	4279 (3532)
- 3.8 kW	1070	453 (355)	2765 (2148)	4288 (3573)
- 6.4 kW	1350	605 (473)	2439 (1895)	4394 (3718)
- 5.3 kW	1530	605 (473)	2428 (1886)	4563 (3889)
- 6.8 kW	1750	705 (552)	1821 (1414)	4276 (3716)

[1] Fuel price inflation rates (derived in Appendix C) are: gas, 9.10% p.a.; wood, 9.75% p.a.; off-peak electricity, 11.58% p.a.; household tariff (direct) electricity, 11.56% p.a. For oil, both the median (24.4% p.a.) and low (12.9% p.a.) estimates are used in NPC calculations.

[2] Maintenance costs are assumed to rise at 9.4 per cent per annum.

Figures 6.1 to 6.6: Net Present costs of living area heating, over a ten-year period, for six case studies

Key to figures:

Cost of depreciation
of initial installation :

Maintenance cost :

Cost of fuel/electricity :

Net Present cost of oil
heating at low (12.9%)
oil price inflation rate : - - -

For definitions of case studies, refer to Table 5.1

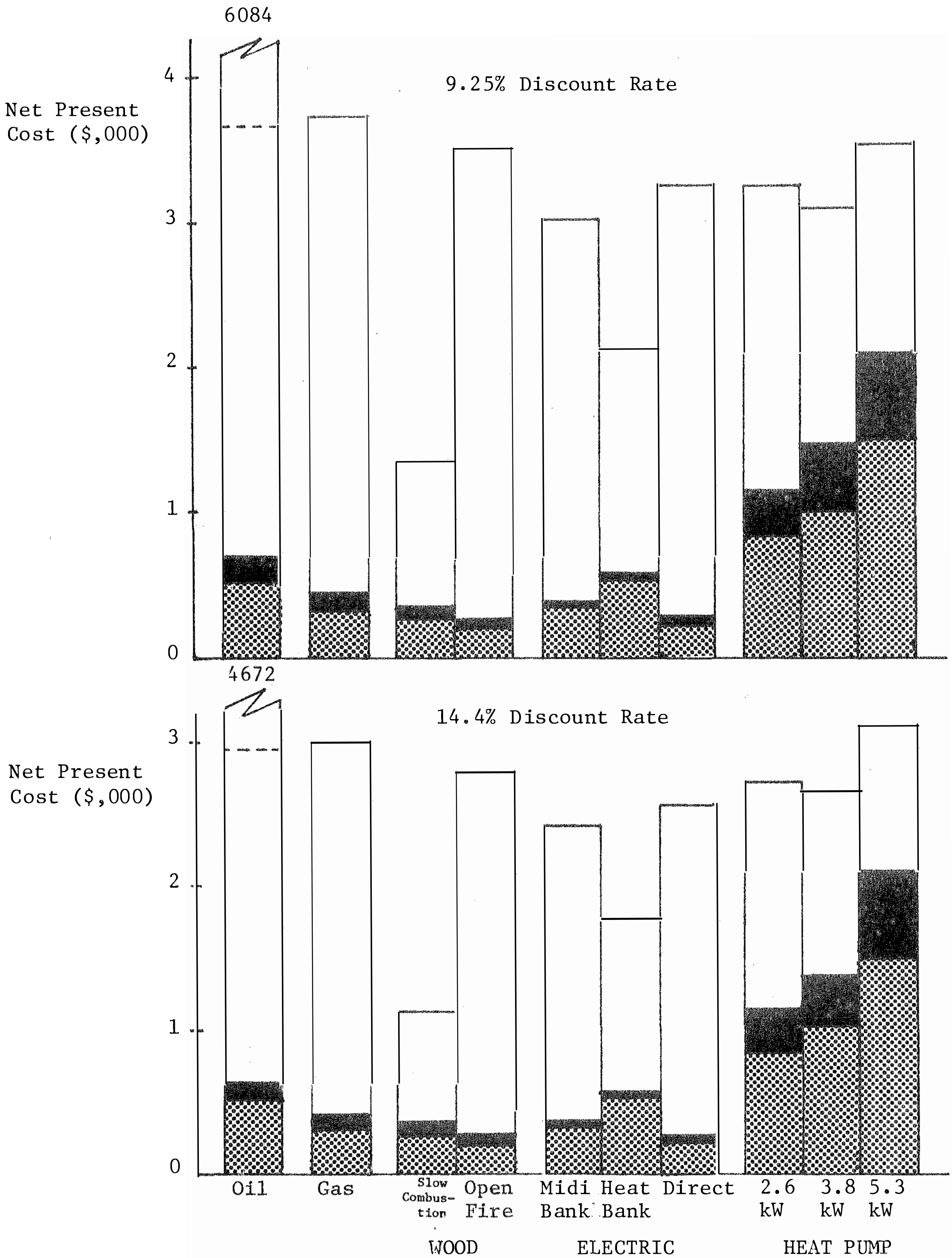

Figure 6.1: Net Present Cost of Heating, Case A: Day-and evening heating, brick veneer house with insulated ceiling

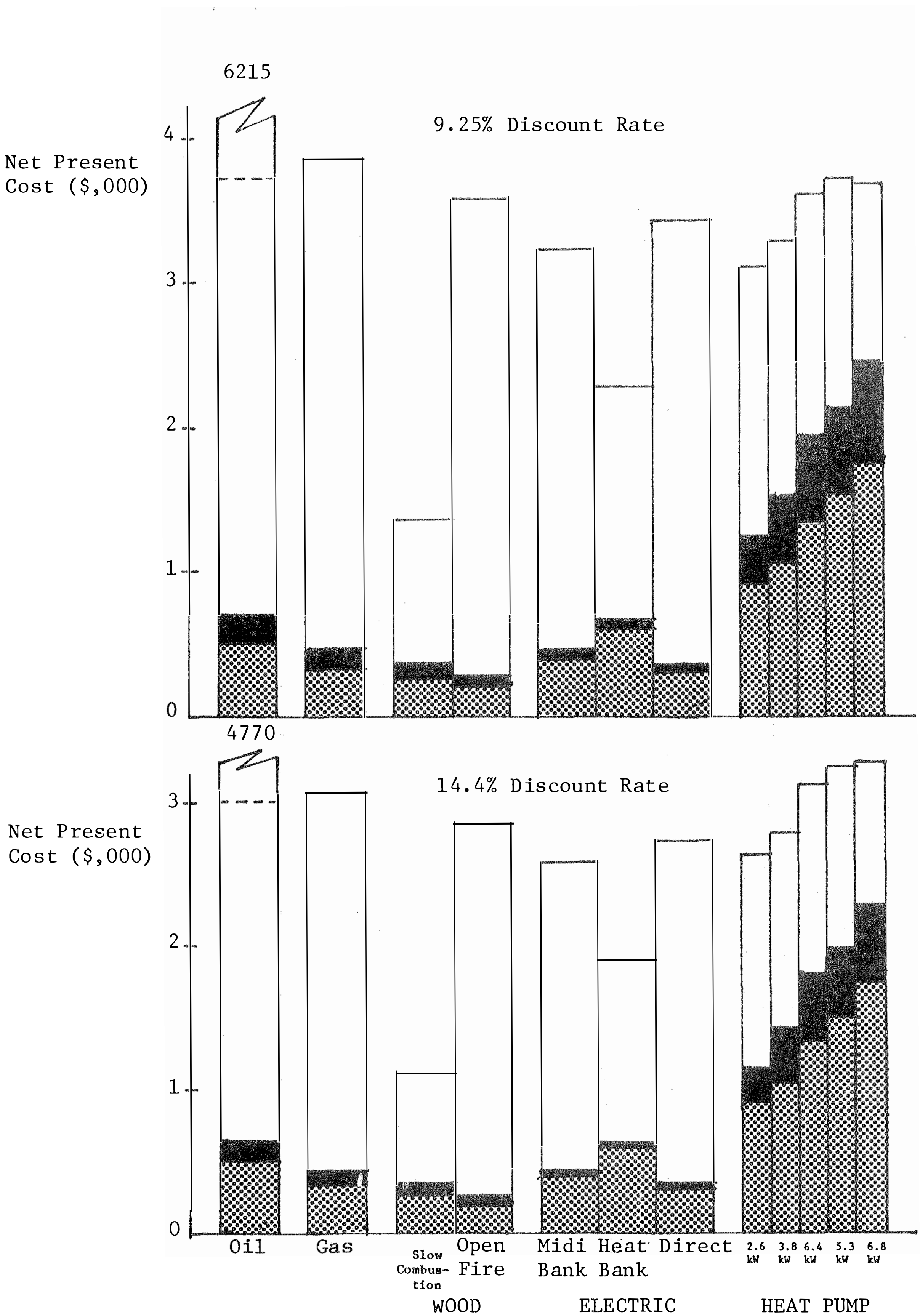

Figure 6.2: Net Present Cost of Heating, Case B: Day-and-evening heating, block veneer house with insulated ceiling

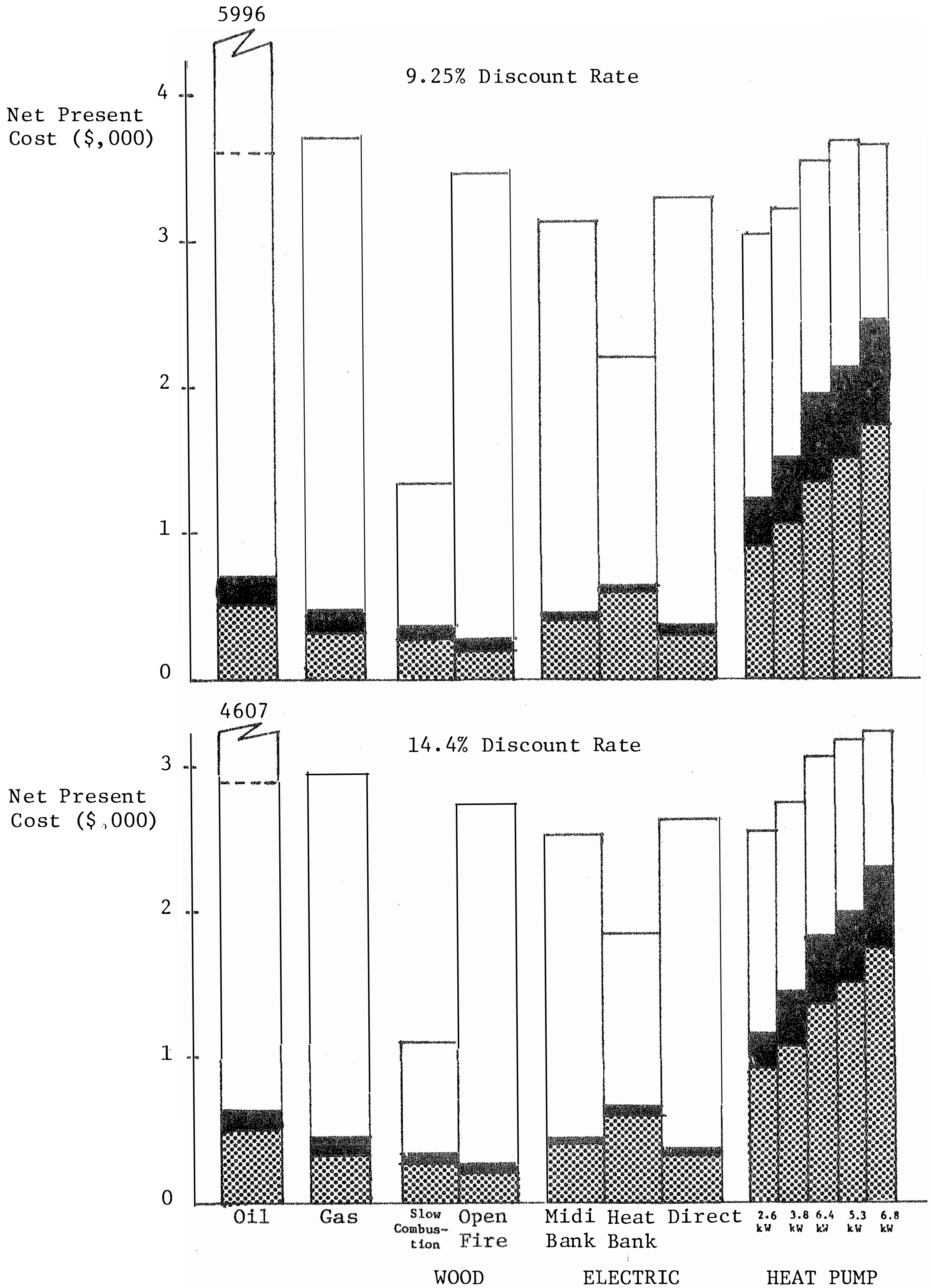

Figure 6.3: Net Present Cost of Heating, Case C: Day-and-evening heating, weatherboard house with insulated ceiling

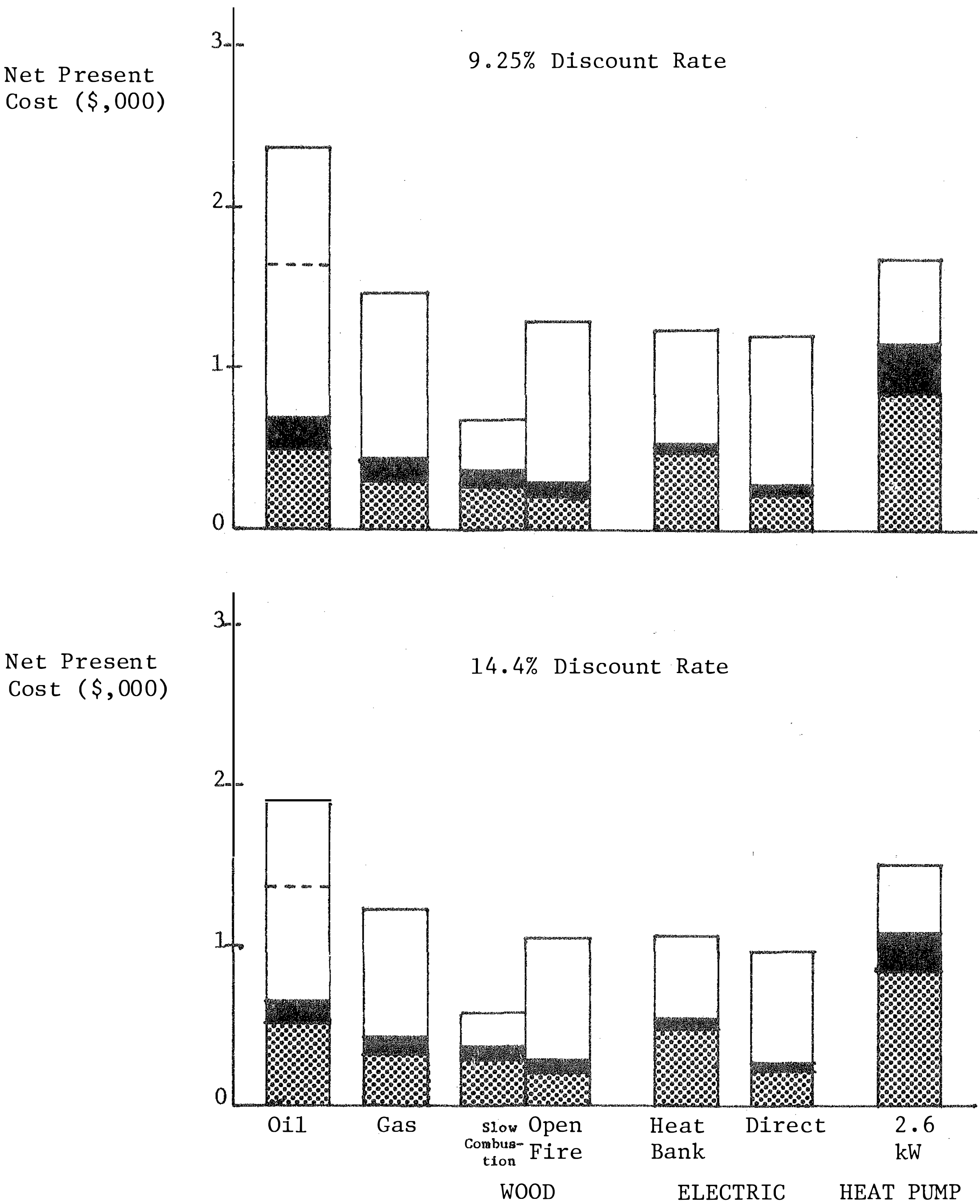

Figure 6.4: Net Present Cost of Heating, Case D: Evening-only heating, brick veneer house with insulated ceiling

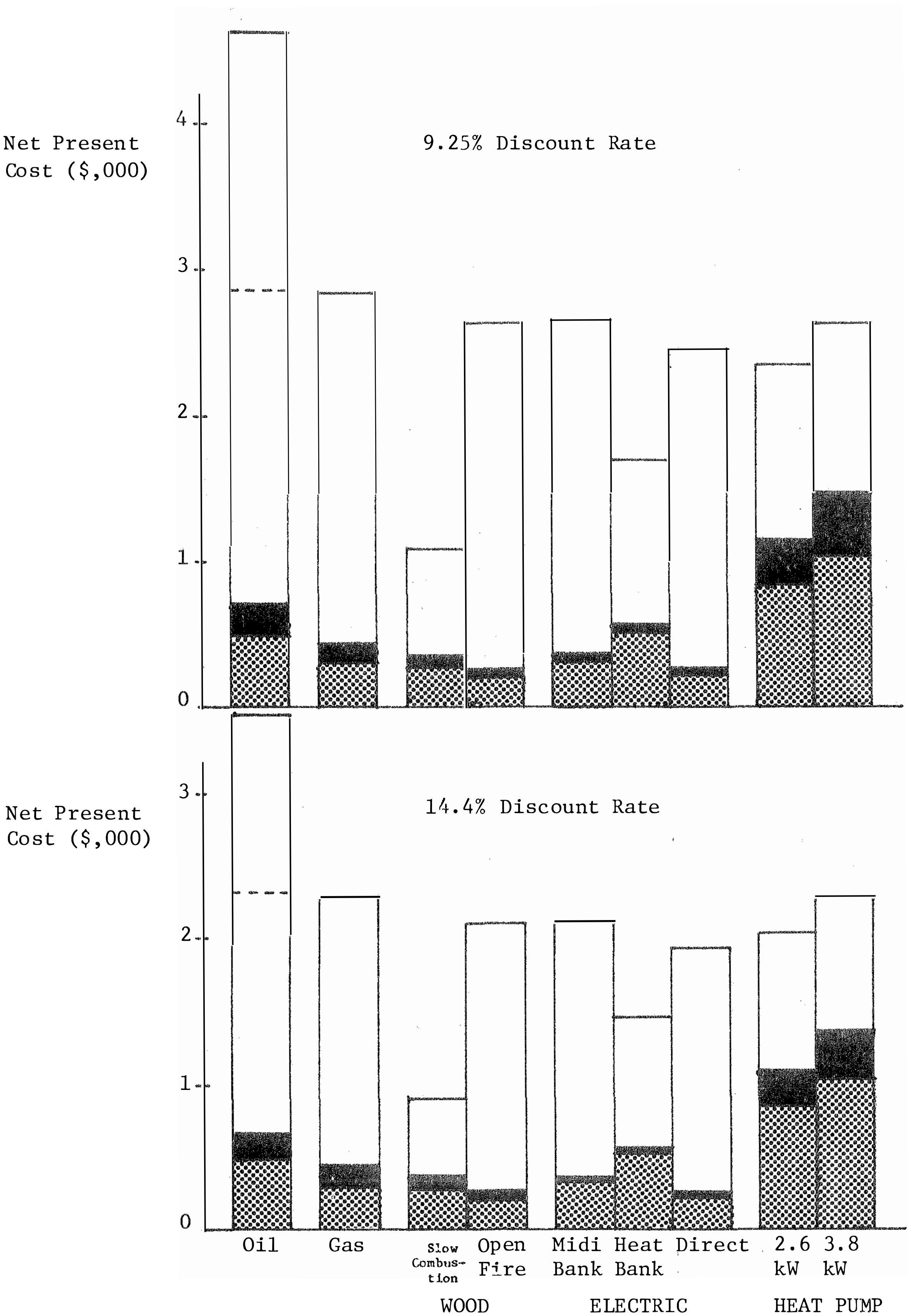

Figure 6.5: Net Present Cost of Heating, Case E: Day-and-evening heating, brick veneer house with insulated ceiling and walls

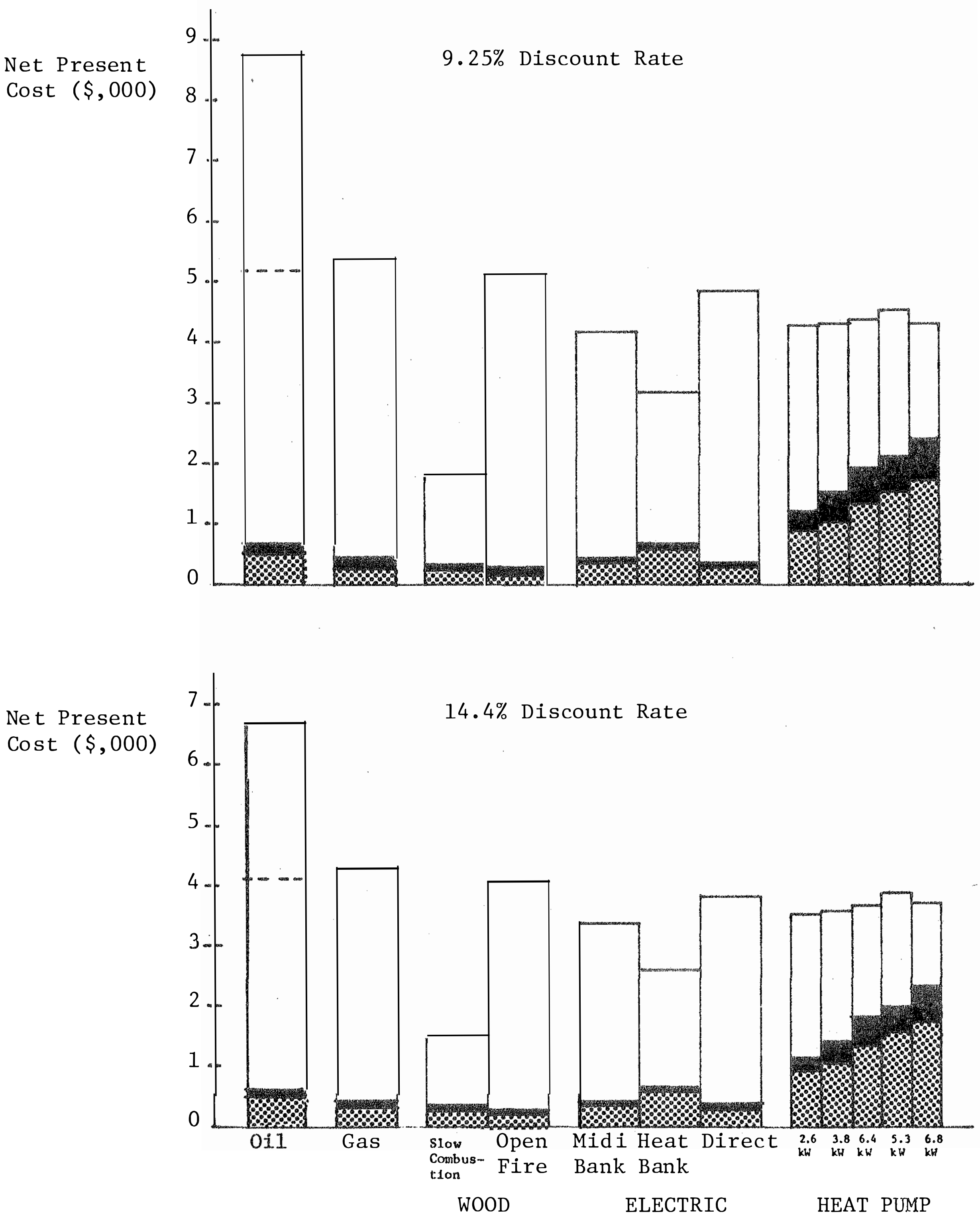

Figure 6.6: Net Present Cost of Heating, Case F: Day-and-evening heating, uninsulated weatherboard house

In the uninsulated house (case F) heat pumps have lower NPC's than direct electric heaters. The only qualification required here is that the 5.3 kW model and the 6.8 kW model (with normal amounts of ducting) have slightly higher NPC's when the higher discount rate is used.

In the houses with insulated ceilings, only the two smallest heat pumps are economic when compared with direct electric heating. The 3.8 kW heat pump has a lower NPC in all cases, but only at the low (9.25 per cent per annum) discount rate. The 2.6 kW heat pump is economic in all three houses at the lower discount rate, and in the block veneer and weatherboard houses at the higher rate. In the brick veneer house, at the higher interest rate, the NPC of the 2.6 kW heat pump is 5% greater than that of direct electric heating.

In the fully-insulated house (case E), only the 2.6 kW heat pump is economic, and then only at the lower discount rate.

Economics for Locations other than Hobart

The relative economics of heat pumps and direct electric heating can be estimated for other locations by applying the climatic adjustment factors derived in Chapter Three (using the most economical heat pump in each case). For each heat pump application, the critical value of the climatic adjustment factor is given by the expression

$$\text{critical value} = \frac{\text{NPV of fixed costs}}{\text{NPV of running cost savings}}$$

where the NPV's refer to the difference in costs between direct electric and heat pump systems. Heat pumps will be economically favoured when the climatic adjustment factor falls above the critical value.

For example, in case B, the most economical heat pump is the 2.6 kW model, with a depreciated initial cost of \$920, net maintenance costs (at the 9.25 per cent discount rate) of \$302, and net running costs of \$1888. The competing direct electric heating system has a depreciated initial cost of \$326, net maintenance costs of \$50, and net running costs of \$3046. Thus, (at the 9.25 per cent discount rate) the NPV of fixed costs is $\$[(920+302) - (326+50)]$, or \$846, whilst the NPV of running cost savings is \$1158. So the critical value in this case is 0.73.

Table 6.4: Economic applications for heat pumps, selected locations within Tasmania[1] (heating to living zone, 20°C ± 1°C).

Period of heating:	day-and-evening[2]					evening[3]
Insulation	none	ceiling[4]			full[5]	ceiling[4]
Case Critical value[9]	F[6] 0.61 (0.73)	A[7] 0.92 (1.09)	B[8] 0.73 (0.88)	C[6] 0.76 (0.92)	E[7] 0.94 (1.13)	D[7] 2.25 (2.72)
LOCATION[10]						
Hobart (1.00)	√√	√	√√	√√	√	X
Launceston (0.81)	√√	X	√	√	X	X
Devonport (0.95)	√√	√	√√	√√	√	X
Burnie (1.03)	√√	√	√√	√√	√	X
Queenstown (1.09)	√√	√√	√√	√√	√	X
New Norfolk (0.93)	√√	√	√√	√√	X	X
Oatlands (1.11)	√√	√√	√√	√√	√	X

1 Explanation of symbols: √√ - economic compared with direct electric heating at 9.25 per cent and 14.4 per cent discount rates.
√ - economic compared with direct electric heating at 9.25 per cent discount rate only.
X - not economic at either discount rate.

2 Heating period 7 a.m. - 11 p.m.

3 Heating period 5 p.m. - 11 p.m.

4 100 mm loose-fill fibreglass + RFL, or 75 mm mineral wool batts.

5 As in 4, plus RFL in walls.

6 Weatherboard house.

7 Brick veneer house.

8 Concrete block veneer house.

9 Critical value of climatic adjustment factor at 9.25 per cent discount rate; critical value at 14.4 per cent interest rate shown in brackets.

10 Values of climatic adjustment factor shown in brackets.

Critical values have been calculated for each case to indicate which heat pump applications are economic at selected locations in Tasmania (Table 6.4).

6.4 A PLACE IN THE MARKET

In the applications previously described (Table 6.4), heat pumps can compete economically with oil, gas, direct electric heating and open fires, but not with slow combustion heaters or off-peak heating. To place this in perspective, the economic factors should be related to the shares of the market held by these heaters (Table 6.5).

Table 6.5: Source of energy for heating main living area.

Source of energy used for heating main living area:	Open fire	Solid fuel	Oil	Kerosene	Electrical o/p	Electrical direct	Gas
June 1977	25.5	10.5	40.4	3.4	6.5	9.2	1.0
June 1978	24.2	10.3	40.4	3.4	7.1	10.2	0.6

[Source: A.B.S. (1978)]

The market for slow combustion (solid fuel) heaters is limited primarily by the inconvenience of constant handling of wood. Off-peak heaters accounted for only 7.1 per cent of installed heaters in 1978. Of the remainder, the largest portion is occupied by oil heaters. As oil prices rise, most of this 40 per cent of heaters is expected to be replaced by other forms of heating, including heat pumps. It is likely that off-peak heating will take a large share of the replacement market.

The heat pump systems found to be economic are through-the-wall types, with substantially less than the whole peak load being supplied by the heat pump. In the calculations, it has been assumed that auxiliary heating is supplied from plug-in electric heaters at $50 per rated kilowatt. Such systems would offer no improvement in convenience (or economy) over off-peak electric systems. Supplementary heaters are available for most heat pumps (at prices around $60 for 3 kW) to maintain output at low outdoor temperatures. Auxiliary heating, designed to boost output under high load conditions, could be included in the same manner. With the high airflow rates used in heat pump operations, it is unlikely that moderate increases in heat output will cause overheating problems.

To combine economy and convenience, these booster elements should be thermostatically controlled, to cut in only when the heat pump is unable to maintain the comfort temperature unaided.

A thermostatically-controlled heat pump installation, properly sized and with adequate auxiliary heating, could compete on convenience and quality of heating (i.e. heat distribution, etc.) with slow combustion and electric off-peak heating, and also,on economy, with other forms of heating.

6.5 ENERGY-CONSERVING INVESTMENTS: HEAT PUMPS AND INSULATION

The annual heating energy requirements of an uninsulated house can be reduced by approximately 34 per cent by installing ceiling insulation, for a net investment (allowing for savings on installed heating capacity) of around $200.

A small (2.6 kW) heat pump can reduce the heating energy requirement by a similar amount (31 per cent) for a net investment of around $650.

To obtain a greater energy saving, wall insulation can also be used. This leads to a reduction of 51 per cent (including the reduction due to ceiling insulation) at a net cost of around $300, if RFL can be used, or $400-$700 using fibreglass batts or urea foam. A similar saving (45 per cent) can be obtained using a larger heat pump (6.4 kW) at a net cost of over $900.

Hence, insulation represents a better energy-conservation investment than heat pumps.

However, it is unusual for wall insulation to be added as a retro-fit to a house, and rare for further insulation to be used. Also, it must be remembered that no energy is conserved by insulation unless a heater of some type is used. Heat pump systems can be used, in conjunction with insulation, to improve the energy efficiency of heating. In all cases of day-and-evening heating, there are cost-effective heat pump systems with lower energy requirements than the normal requirements of a fully-insulated house. In the extreme case, a heat pump can reduce the heating energy requirements of a fully-insulated house by over 40 per cent.

In terms of energy conservation, then, heat pumps are less *cost-effective* than insulation, but more effective in absolute terms.

6.6 SOURCES OF UNCERTAINTY IN THE ECONOMIC COMPARISON

(i) Price of heating oil

It has been assumed that the price of heating oil will increase at a relatively high rate of 24.4 per cent. At this rate, oil heating will be hopelessly uneconomic in the heating applications studied. It is possible that Australia's oil prices will tend to stabilise on reaching world parity, and that world prices will stabilise when the demand has been reduced. In this case, it may be appropriate to use the lower price increase rate of 12.9 per cent. Even at this rate (as shown by the dotted lines in Figures 6.1 - 6.6) oil no longer represents cheap heating, being more expensive in all cases than direct electric heating.

(ii) Price of L.P. Gas

Future prices of L.P. gas depend very much on Government policy. At present, it is about 25 per cent more expensive, for heating, than electricity at the household tariff. If the price of L.P. gas becomes tied to oil prices, then gas will become a very expensive form of heating. If, on the other hand, gas prices are reduced to levels comparable to those in mainland states, gas heating may become competitive with direct electric heating.

(iii) Open fireplaces

Estimates of the efficiencies of open fireplaces range from 10 per cent to 20 per cent, but the author has found no evidence on which these estimates could be based. An efficiency of 15 per cent has been assumed, but costs could be up to 50 per cent higher or 30 per cent lower, at respective efficiences of 10 per cent and 20 per cent. Since the open fire heats predominantly by radiant heating, over a restricted area within approximately ± 45^{o} of the fireplace, it is not really suitable for space heating. Estimation of heating costs is further complicated by a lack of information on the heat output of the open fireplace. Hence, the estimates of NPC relating to open fires must be regarded as tentative, at best.

(iv) Off-peak electric heating

The off-peak tariff appears tied to a level of approximately 45 per cent of the marginal household tariff. The relevance of an off-peak tariff in a hydro-electric system - in which generators can be taken into or out of the grid in a matter of seconds - remains largely unexplained. The major problems in Tasmania's electricity system, both in the past and in the future, are related to total annual demand rather than supply of peak loads. Thus, it is difficult to justify a low off-peak heating tariff, particularly if it causes a significant increase in total energy consumption.

The real reason for the off-peak tariff in Tasmania seems to be as a means of selling electricity in a competitive heating market. With escalating prices of the major competitor (oil) this reason is no longer valid, but people who have invested in off-peak heating cannot be expected to pay normal prices for their off-peak electricity.

A large proportion of the 40.4 per cent of homes presently using oil heaters is likely to switch to electrical heating in the coming few years. If three-quarters of them replace their oil heating with the same amount of electric heating, total annual electricity demand will be boosted by around 8 per cent, with much of this increase being bought at low off-peak rates.

With increased electricity demand from other sources already expected to account for the full hydro-electric generating capacity, this further demand will probably be supplied at high marginal cost from the oil- (or, in the future, coal-) fired Bell Bay power station. Thus, increased demand for off-peak heating could result in a net cost to the Hydro-Electric Commission. The most acceptable strategy to avoid this would be to discourage the installation of off-peak heating. A step was made in this direction during 1978, when the installation of off-peak power points was disallowed. Off-peak appliances must now be permanently wired-in.

If the availability of off-peak heating becomes restricted, heat pumps will become the most cost-effective convenient means of space-heating in many applications.

(v) Direct electric heating

Since the heat pumps described operate on the same tariff as direct electric heaters, their economics are relatively unaffected by variations in the price inflation rate of the household electrical tariff. However, because heat pumps use less electricity, they will become more (or less) cost-effective if the household electrical tariff increases (or decreases) in real terms.

6.7 IMPLICATIONS OF THIS STUDY FOR HYDRO-ELECTRIC COMMISSION POLICY

The results of the NPC calculations indicate quite clearly that in all forms of electric heating, the major financial interest is held by the body (in this case, the Hydro-Electric Commission) supplying the electricity.

The swing away from oil heating may result in an undesirable growth in annual electricity demand, due to an increased use of electric heating. The only type of electric heater which reduces both peak and annual electrical loads is the heat pump. Further, the revenue from heat pumps is comparable to that obtainable from off-peak heating, with considerable savings in electricity requirements. Thus, it is in the interests of the Hydro-Electric Commission to encourage the use of heat pumps.

One means of achieving this would be to offer loans to enable customers to install heat pump heating. Since heat pumps can be cost-effective at the bank interest rate, this approach would provide a satisfactory result for both the lender and the customer. Loans could be offered either by the Commission itself, or by banks, for heat pump installations approved as economic by the Commission.

Perhaps a more important step would be to institute an Assured Heat Pump Service Programme, as has been done by the Alabama Power Company. The main purpose of this programme would be to boost customer confidence by ensuring (by means of reasonably-priced service contracts) a high standard of maintenance and reliability.

The need for such a scheme is evidenced by the fact that at least one major Australian manufacturer has no service representative in Tasmania.

6.8 IMPLICATIONS OF THIS STUDY FOR THE TASMANIAN HOUSING DIVISION

The Tasmanian Housing Division, from its inception, has always recognised the need for heating in Tasmanian homes. More recently, it has acknowledged the need for energy conservation by installing ceiling insulation in all its newly-completed homes.

However, things are still far from ideal. People in homes built by the Division cannot afford expensive heating bills, yet in 1979 nearly all its detached houses are heated either by open fireplaces or oil heaters — two of the most expensive forms of heating!

The economic analysis indicates quite clearly that the cheapest form of heating is the slow combustion wood heater, which can be installed at a lower cost than either an oil heater or a fireplace.

If the tenant finds the inconvenience of tending a slow combustion heater unacceptable, then the next economic choice is the electric heat bank. Should the price of off-peak electricity rise in relation to the household tariff, or the availability of off-peak heating become restricted, then heat pumps may be chosen in preference to heat banks. The same result will occur if the price of heat pumps falls or higher COP's are obtained.

At present, the economic heating choices are slow combustion stoves and electric heat banks. The Housing Division would be well advised to keep a watchful eye on new developments in heat pumps.

One problem with the use of heat banks or through-the-wall heat pumps in the two Housing Division designs used here is the lack of communication between living room and kitchen. It has been assumed that heat passes between them via the hallway. A more satisfactory solution would be to have a direct connecting door, or to use a heater mounted in the dividing wall and fitted with a back vent (e.g. a split-system console heat pump). Future designs will need to be drawn up to best suit the type of heater that is intended to be used.

6.9 GENERAL CONCLUSIONS

Heat pumps are an energy-efficient means of space heating. They are economic for heating the living areas of Tasmanian homes when heating is required during both the day and evening.

The most cost-effective type of heat pump is the through-the-wall unit, and the economical size, for design heating loads up to 7 kW, is approximately half the design load, with the remainder of the peak load supplied by electric resistance heating. At higher design loads, larger and more efficient heat pumps, with capacities around two-thirds of the design capacity, become economic.

Energy savings obtained using these systems range from 30 per cent to over 50 per cent, and when combined with insulation can be as high as 70 per cent.

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APPENDIX A: EFFICIENCY OF HEATERS

ELECTRIC RESISTANCE HEATING:

Electric resistance heaters are assumed to be 100 per cent efficient.

OIL HEATERS:

Bonne, Janssen and Torborg (1977) have calculated seasonal efficiencies for a number of oil heating systems. Calculated efficiencies range from 55 per cent to 75 per cent, being typically in the range 65-70 per cent.

It is generally assumed that operating oil heaters at low load levels reduces their efficiency. Low load levels occur more frequently in over-sized heaters (i.e. heaters with maximum outputs above the design heating load of the building), and corrections to seasonal efficiency can be made according to the amount of over-capacity. Correction factors have been calculated by Bonne, Janssen and Torborg (1977) and ASHRAE (1973). The corresponding efficiencies are shown in the figure below.

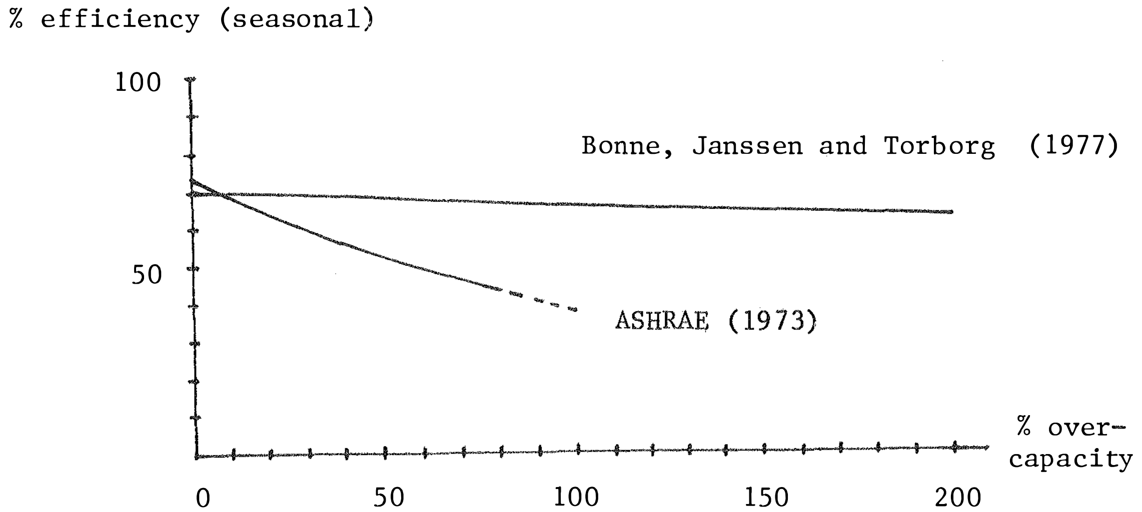

Figure A.1: Effect of over-capacity on efficiency of oil heaters.

The difference between these two sets of results indicates that the correction factor depends very much on the assumptions made. The ASHRAE corrections are empirical corrections applied in general to oil- and gas-fired central heating systems. Those of Bonne, Janssen and Torborg refer to specific oil-fired central heating systems. In both cases, a significant contribution to inefficiency at low load levels comes from stack losses, and the heat absorbed by the furnace itself during warm-up. These losses are minimal in the console-type heaters installed *within* the heated areas of Tasmanian homes. Thus, the validity of these corrections is questionable.

For the purposes of this report, oil heaters are taken to have a seasonal efficiency of 70 per cent (i.e. inefficiencies due to over-sizing are ignored).

GAS HEATERS:

Chi and Kelly (1978) have estimated seasonal efficiencies for a number of gas-fired furnace systems using indoor air for combustion. Three of these systems, without pilot lights, have seasonal and annual efficiencies of 68.6, 74.1 and 74.8 per cent.

Radiant gas heaters with automatic ignition, and using indoor air for combustion, will have similar efficiencies. The efficiency of these heaters is taken as the mean of the three figures given above, i.e. 72.5 per cent.

Modern gas heaters consist of a number of radiant heating tiles. Each tile is normally operated at full output, and the total output is varied by changing the total number of tiles in operation. Hence, there is no need to adjust efficiencies for over-capacity.

WOOD HEATERS:

Assumed efficiencies of open fires range from 10 per cent to 20 per cent. A median value of 15 per cent is assumed here. At 10 per cent efficiency, costs will be approximately 30 per cent greater than those quoted, while at 20 per cent efficiency they will be 20-25 per cent lower.

Shelton *et al.* (1978) have tested the efficiency of a slow combustion

(Jøtul model 602) wood-burning stove, and found that efficiency depends primarily on the moisture content of the wood. Efficiency of combustion was highest at 25 per cent moisture content, but heat transfer decreased constantly with moisture content. Overall heating efficiency increased from 54 per cent (zero moisture content) to 63 per cent (15 per cent moisture), and then declined to 50 per cent (30 per cent moisture).

The tested stove did not appear to have a flue damper, and transferred its heat to the room by natural convection and radiation (from its cast-iron surfaces). Slow combustion space heaters of the type used in Tasmania are equipped with flue damping devices and fan-assisted convection heat exchangers. Hence, they are likely to have higher efficiencies than those of the machine tested. Seasonal efficiencies will be lower than instantaneous efficiencies, due to the heat required for initial heating of the firebase and chimney, and the heat of combustion that occurs late at night when the living room has been vacated. The seasonal efficiency of a slow combustion heater is taken to be 60 per cent.

Efficiencies are summarised in Table A.1.

Table A.1: Efficiencies of heaters.

Heater Type	Efficiency (%)
Electric resistance	100
Oil	70
Gas	72.5
Open fire	15
Slow combustion	60

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APPENDIX B: ENERGY CONTENTS AND PRICES OF FUELS

The energy contents, prices, and costs of fuels are summarised in Table B.1.

Table B.1: Energy contents, prices and costs of fuels.

Fuel type	Retail price[1]	Energy content	Cost per unit energy $/GJ	Heating efficiency[2] %	Heating cost[3] $/GJ
HEATING OIL	17.58 ¢/litre	37.6 MJ/litre[5]	4.68	70	6.68
L.P. GAS	$17 per 45 kg cylinder	49.75 MJ/kg[6]	7.59	72.5	10.47
WOOD					
- 9" (slow combustion)	$28/tonne	15.6 GJ/tonne[7]	1.80	60	2.99
- 18" (open fire)	$23/tonne	"	1.47	15	9.83
ELECTRICITY					
- Household tariff (marginal tariff, direct heating)	3.14 ¢/unit[4]	3.6 MJ/kWh	8.72	100	8.72
- Off-peak tariff (storage heating)					
First 2000 units:	1.52 ¢/unit[4]	"	4.23	100	4.23
Marginal rate:	1.32 ¢/unit[4]	"	3.68	100	3.68

1 Prices determined for Hobart, June 1979.

2 Efficiencies as derived in Appendix A.

3 Cost of energy, adjusted for heating efficiency.

4 One unit of electricity = 1 kWh; prices include 5 per cent Government surcharge.

5 This is the average energy content of heating oil, as given by the Australian Institute of Petroleum (1976).

6 This is the mean of the energy contents of propane (50 MJ/kg) and butane (49.5 MJ/kg), as given by the Australian Institute of Petroleum (1976).

7 This value has been determined in a bomb calorimeter, from a sample of Tasmanian firewood, by John Sutton of the Centre for Environmental Studies at the University of Tasmania (pers. com.), and compares closely with that (15.4 MJ/kg)

given in a listing of energetic values prepared by the Fuels Branch, Australian Department of Minerals and Energy, for the 1976 conference of the Australian and New Zealand Association for the Advancement of Science (ANZAAS).

REFERENCES:

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APPENDIX C: TRENDS IN FUEL PRICES

C.1: INTRODUCTION

A comparison of the economics of heating systems requires a forecast of fuel prices over the period of comparison (in this case, ten years). Events in the past decade have shown that reliable and accurate forecasting of fuel prices is an impossibility. What is aimed for here is rather a reasonable indication of price trends, based on the price trends recorded in the decade just completed.

In order to improve the reliability of the projections, the forecast will be taken to be the median of projections obtained by three different methods. The methods used are:

(i) mean fuel price inflation rate over ten years (1969-1979);

(ii) mean fuel price inflation rate over five years (1974-1979);

(iii) prices predicted by a regression on annual rates of price increase (1969-1979).

The first two methods are relatively simple. The third method involves taking annual inflation rates over ten years and fitting a straight line to them (see section C.2 for an example of the use of this method). It has an inherent instability, since it predicts a continuous increase or decrease in inflation rates. However, in one case (oil prices) it provides a much closer fit to observed price trends, and shows that in that case a constant inflation rate is statistically improbable.

The following sections describe how price trends have been forecast from the recorded prices listed in Table C.1.

C.2: TRENDS IN OIL PRICES

The regression method has been used to predict prices by finding the line of best fit to a graph of annual price increase rates, for heating

Table C.1: Recorded energy costs (¢/MJ[1]) 1968-1979

Year ending 30 June[2]	Electricity[3] household tariff	Electricity[3] off-peak tariff	Heating oil	Wood	L.P. Gas
1968	0.570	0.200	0.136	0.046	0.50
1969	0.570	0.200	0.139	0.046	0.50
1970	0.570	0.200	0.144	0.052	0.50
1971	0.570	0.200	0.147	0.051	0.50
1972	0.669	0.233	0.150	0.047	0.50
1973	0.669	0.233	0.150	0.048	0.50
1974	0.669	0.233	0.157	0.080	0.50
1975	0.786	0.275	0.184	0.086	0.59
1976	0.925	0.322	0.245	0.098	0.68
1977	0.925	0.322	0.271	0.091[5]	0.68[6]
1978	0.925	0.322	0.330[4]	0.099[5]	0.68[6]
1979	1.156	0.403	0.468[4]	0.108[5]	0.76[6]

1 Prices for purchased energy have been (except where otherwise noted) obtained from Hartley, Jones and Badcock (1978). They do not make any allowance for efficiencies of energy use.

2 Heating oil prices are for June 1st of the year given; all other prices in the table are for June 30th of the year given.

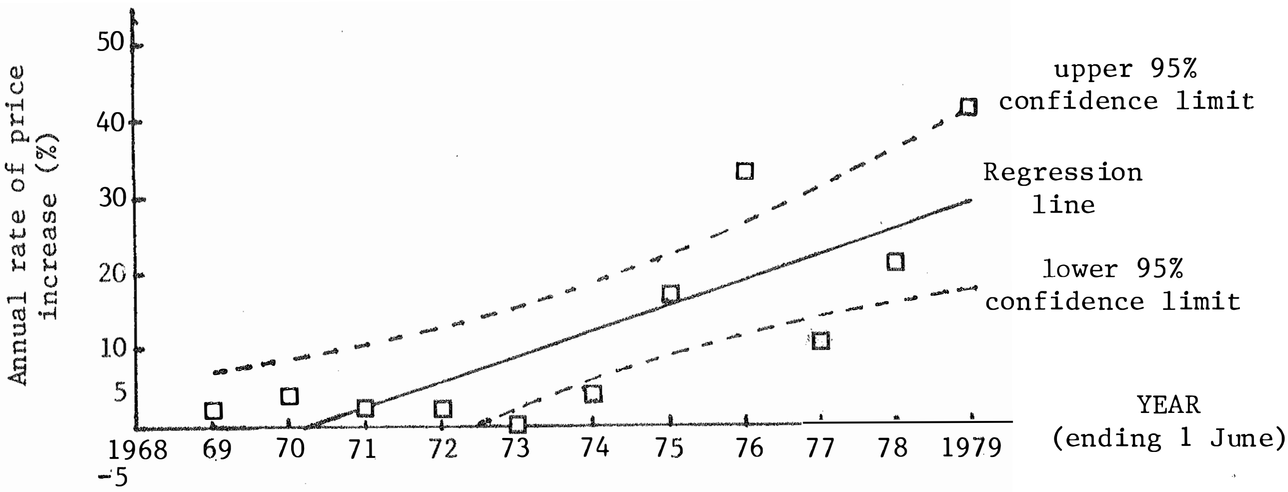
3 Based on Hydro-Electric Commission tariffs, but the plateau scale is not taken into account (e.g. household electricity prices are for the first 300 kWh sold).

4 Based on 1978-1979 heating oil prices quoted by the Shell Company of Australia (Hobart).

5 Based on average spot prices of firewood as advertised in the Hobart *Mercury*.

6 Based on information supplied by Commonwealth Industrial Gases Limited, Hobart.

Figure C.1: Regression line for annual rate of heating oil price increase.



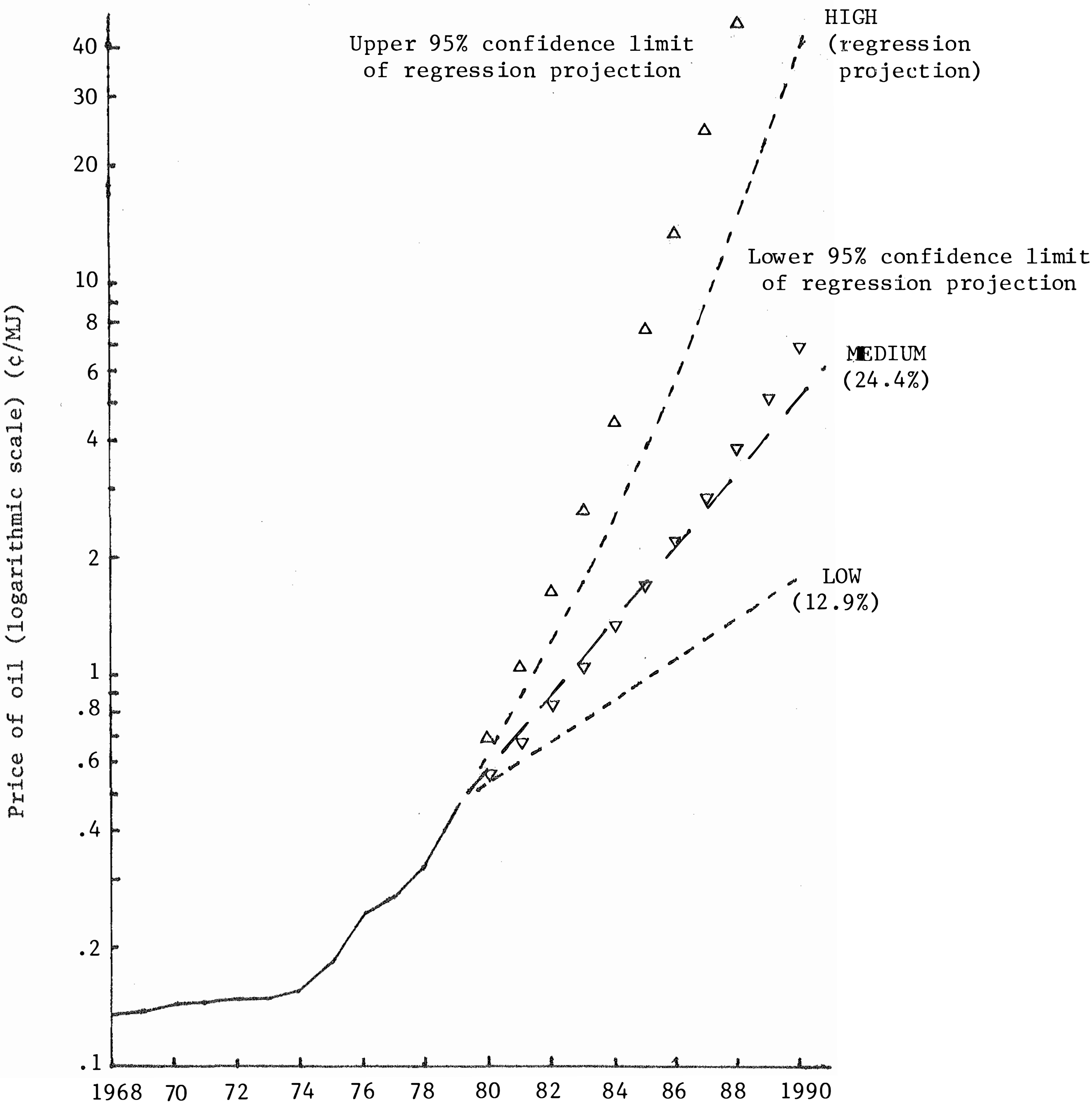


Figure C.2: Projected price of heating oil to year 1990.

oil (Figure C.1). This line has been used to predict annual heating oil price inflation rates to 1990. By applying these inflation rates to the 1979 price, the future prices have been determined (the HIGH estimate of Figure C.2). Ninety five per cent confidence limits have also been determined. The corresponding prices are plotted as points in Figure C.2.

The mean annual price increases (medium and low projections) appear as straight lines in Figure C.2 because of the logarithmic scale used. Only the high (regression) estimate provides an accurate modelling of observed prices from 1969 to 1979. However, the medium (24.4 per cent) estimate lies close to the prices observed between 1974 and 1979.

According to the linear regression analysis, the inflation rate for heating oil is increasing at 3.4 per cent per annum. The lower 95 per cent confidence limit for this estimate is 1.5 per cent, so that the assumption of a constant inflation rate for heating oil is not valid.

Because the future of oil prices is so uncertain, both the median (24.4 per cent) and low (12.9 per cent) inflation rates will be considered. At the high inflation rate, over ten years, oil would become hopelessly uneconomic for heating applications. Hence, it is unnecessary to consider the high rate.

C.3: TRENDS IN L.P. GAS PRICES

The mean inflation rates for L.P. gas have been determined as follows:

1969-1979: 4.45 per cent per annum
1974-1979: 9.10 per cent per annum.

According to the regression method, the inflation rate is rising at 1.0 per cent per year. This rate is not significantly different from zero, at the 5 per cent level. Thus, it is valid to assume a constant inflation rate. The regression method (Figures C.3 and C.4) predicts that the inflation rate for L.P. gas will rise from 9.94 per cent (1980) to 19.70 per cent (1990), giving estimates higher than given by the two mean inflation rates (Figure C.4). The wide confidence limits (Figure C.4) are due to the irregular nature of past price increases, and illustrate the errors involved in extrapolating from past price trends.

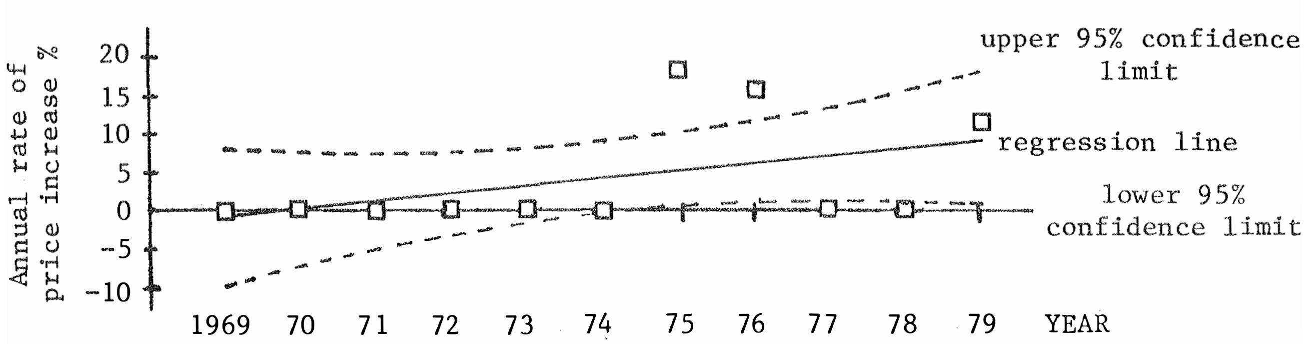


Figure C.3: Inflation rate of L.P. Gas.

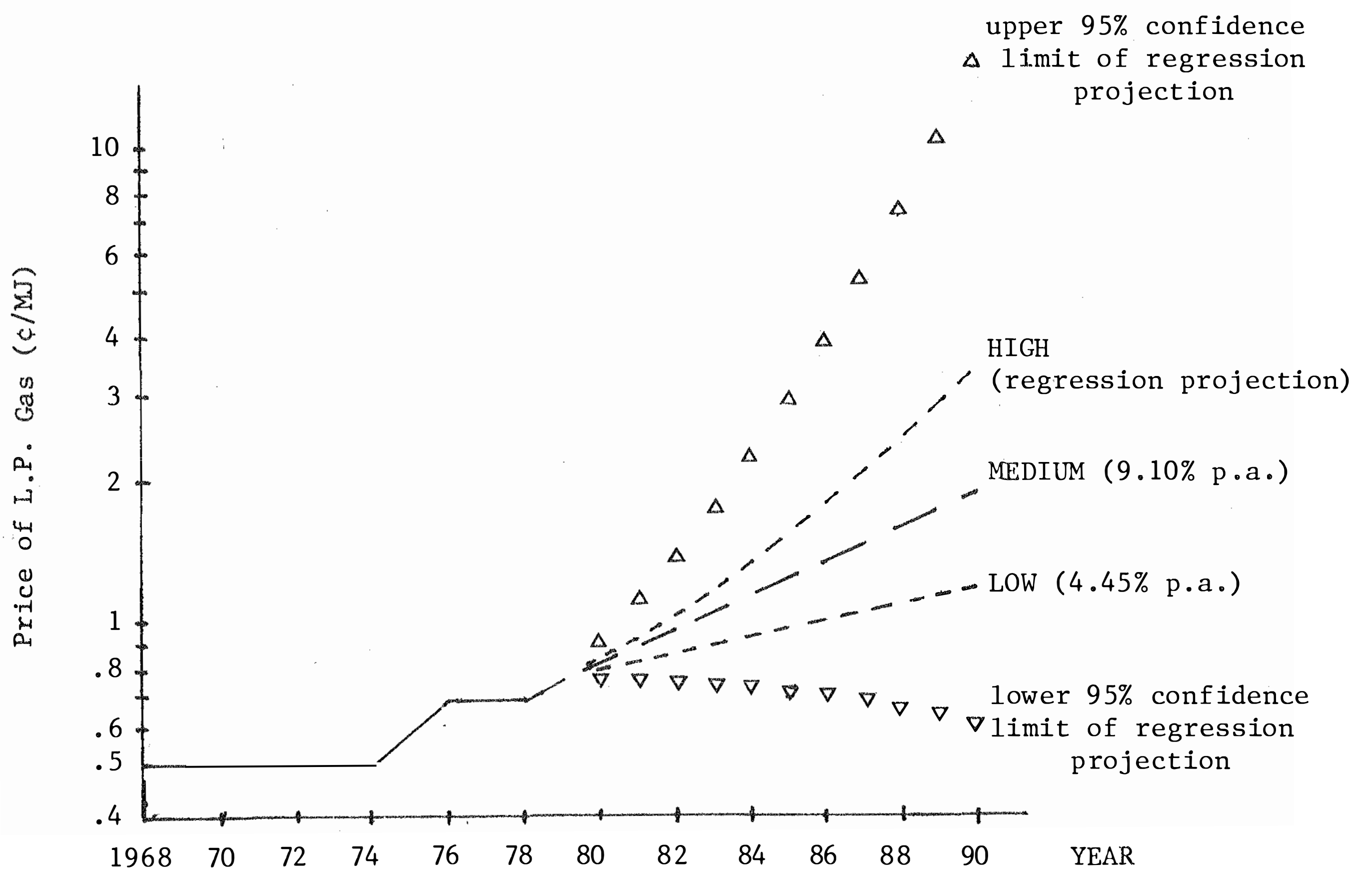


Figure C.4: Recorded and projected prices of L.P. Gas.

The median estimate for the L.P. gas inflation rate is 9.10 per cent per annum.

C.4: TRENDS IN ELECTRICITY PRICES

Mean annual inflation rates for electricity are shown in Table C.2.

Table C.2: Mean annual inflation rates of electricity.

	Household tariff	Off-peak tariff
1969-1979	7.32%	7.26%
1974-1979	11.56%*	11.58%*

* Median inflation rates.

The rate of increase of the inflation rates (1.3 per cent per annum in both cases) is not statistically significant (at the 5 per cent level). Hence, the assumption of constant inflation rates is again valid. Projected electricity prices are depicted in Figures C.5 and C.6. Median inflation rates are the 1974-1979 rates shown in Table C.2.

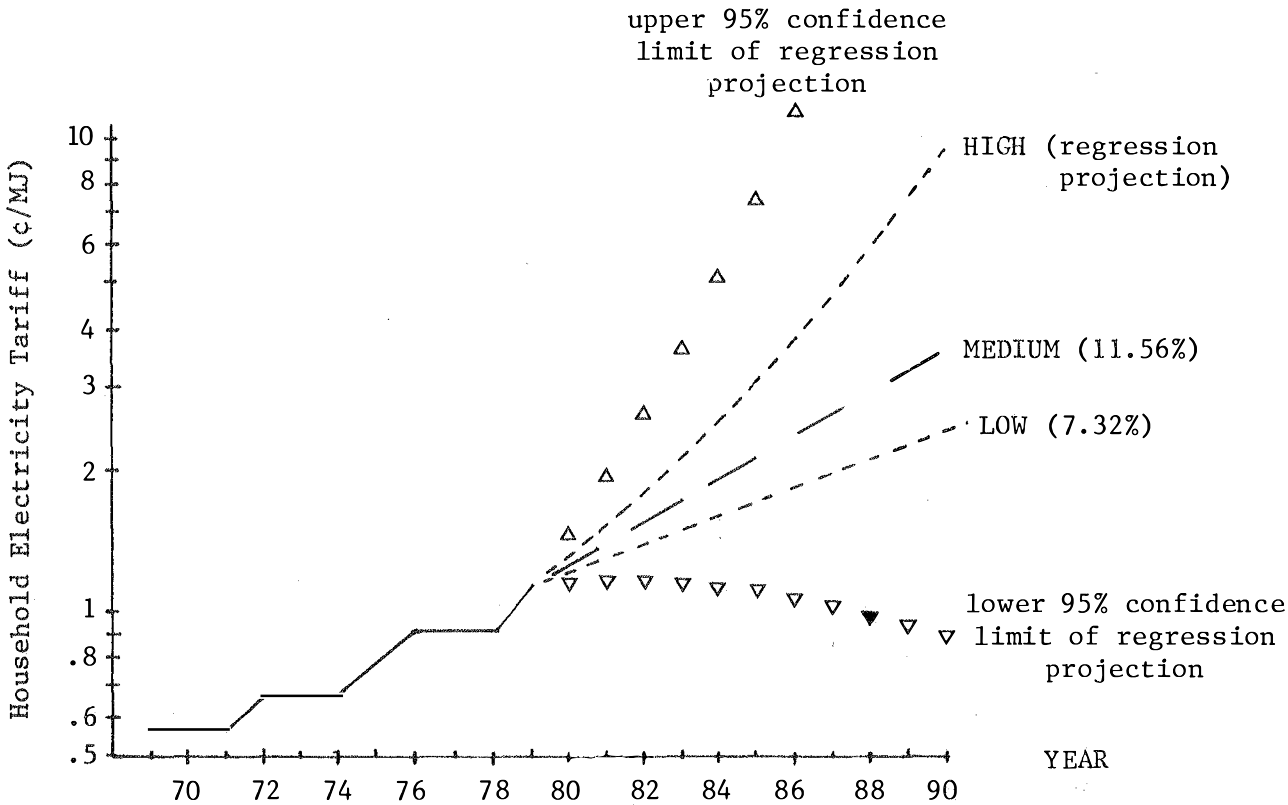
C.5: TRENDS IN FIREWOOD PRICES

In the case of firewood, the median projection is given by the ten-year mean inflation rate (9.75 per cent), which is only slightly higher than the five-year mean inflation rate (8.14 per cent). Due to an apparent discontinuity in prices between 1973 and 1974, the regression projection is high, with very large error limits.

C.6: SUMMARY

Price inflation rates of fuels in the decade 1969-1979 have been

Figure C.5: Household tariff electricity price trends.



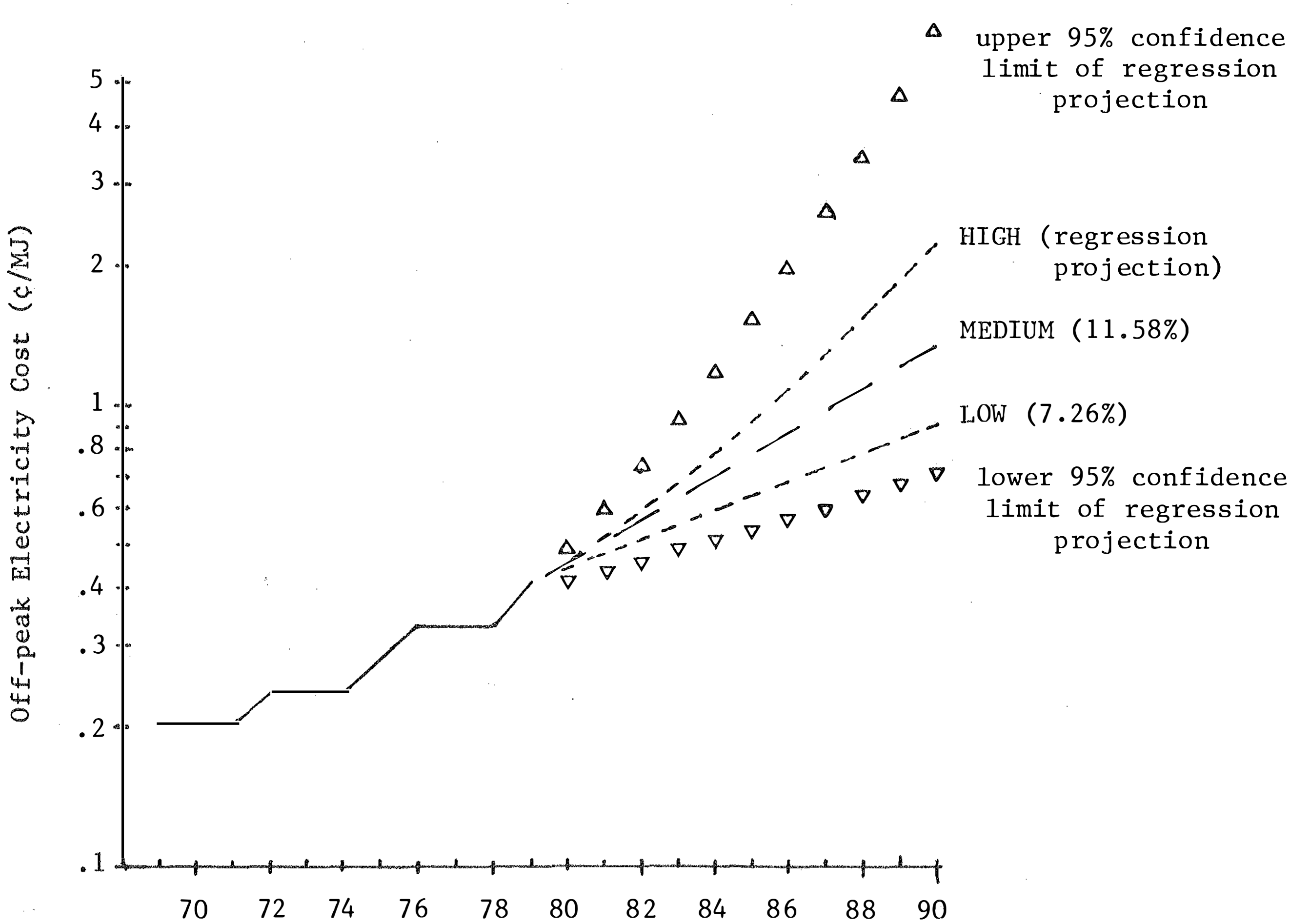


Figure C.6: Off-peak electricity price trends.

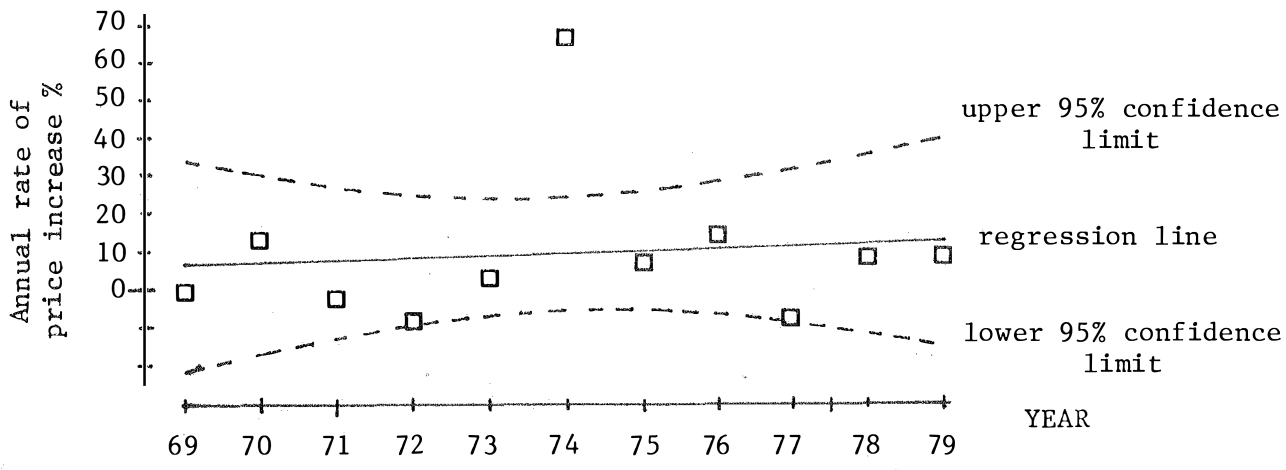


Figure C.7: Inflation rate of firewood.

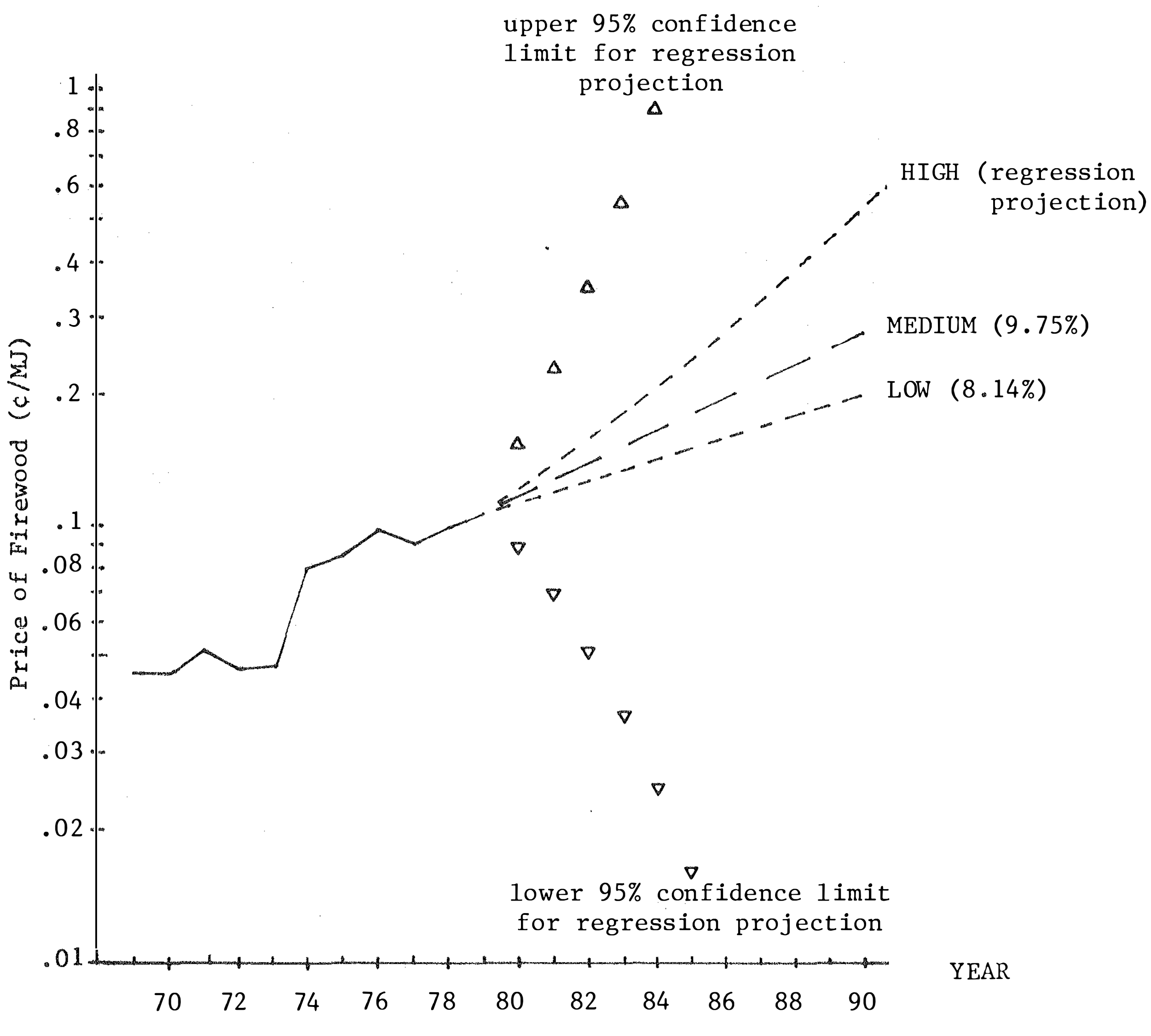


Figure C.8: Firewood price trends.

estimated according to three relatively simple mathematical models. In each case, the model giving the median projected inflation rate in the following decade will be used in the economic analysis (Chapter Six).

Projected inflation rates for fuels are listed in Table C.3.

Table C.3: Projected fuel price inflation rates, 1980-1990.

Fuel	Fuel price inflation rate
Oil	24.4% p.a.[1]
Gas	9.10% p.a.
Wood	9.75% p.a.
Electricity	
- household tariff	11.56% p.a.
- off-peak tariff	11.58% p.a.

[1] Low estimate of inflation rate 12.9 per cent per annum.

REFERENCE:

HARTLEY, M.J., JONES, R. and BADCOCK, R.L., 1978; *Growth and Development of Tasmania's Energy System: A Statistical Analysis of Supply and Demand 1950-1975*; Environmental Studies Working Paper 9, Board of Environmental Studies, University of Tasmania.